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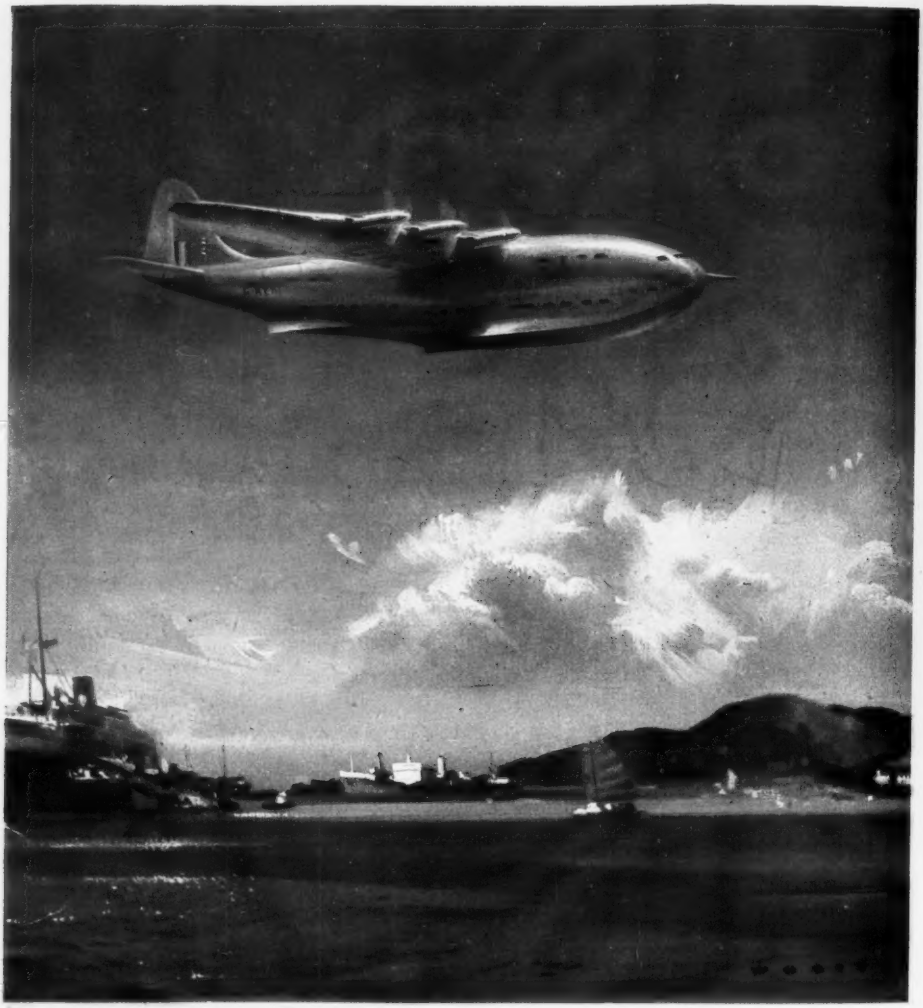
The Journal of
THE ROYAL AERONAUTICAL SOCIETY

MARCH 1946

NO. 423 VOL. 50

SUPREME FOR OCEAN TRAVEL – THE BRITISH FLYING BOAT



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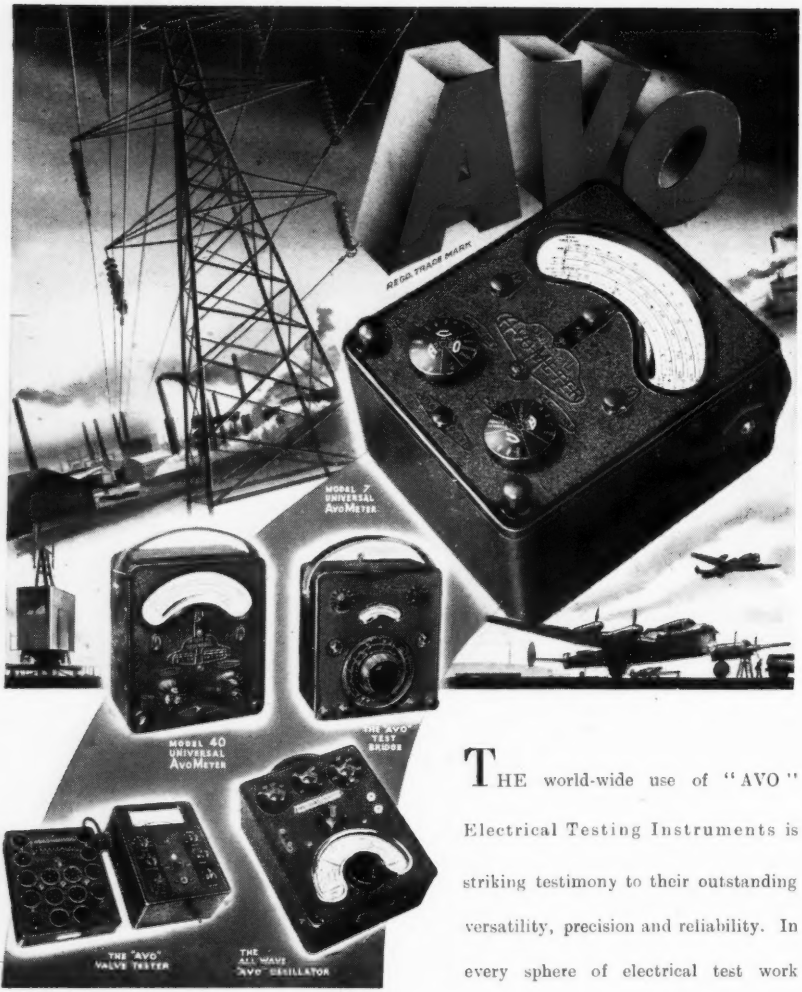

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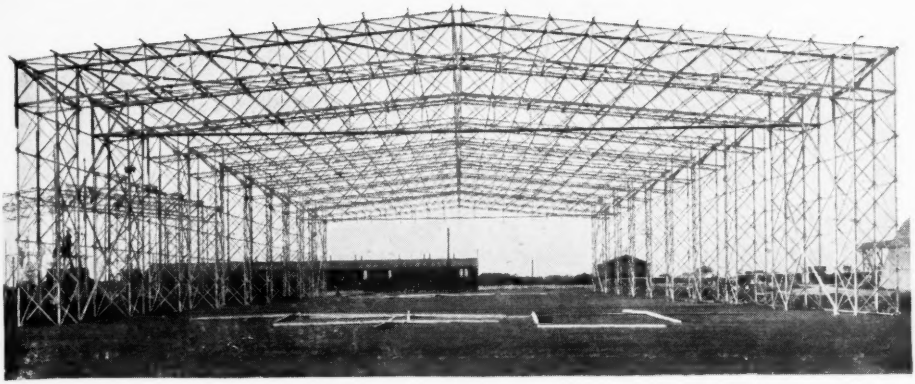

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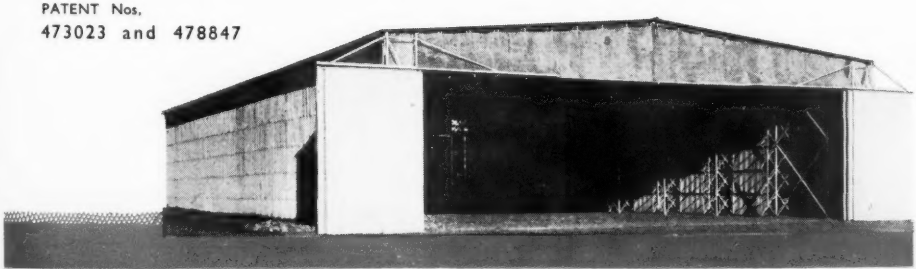

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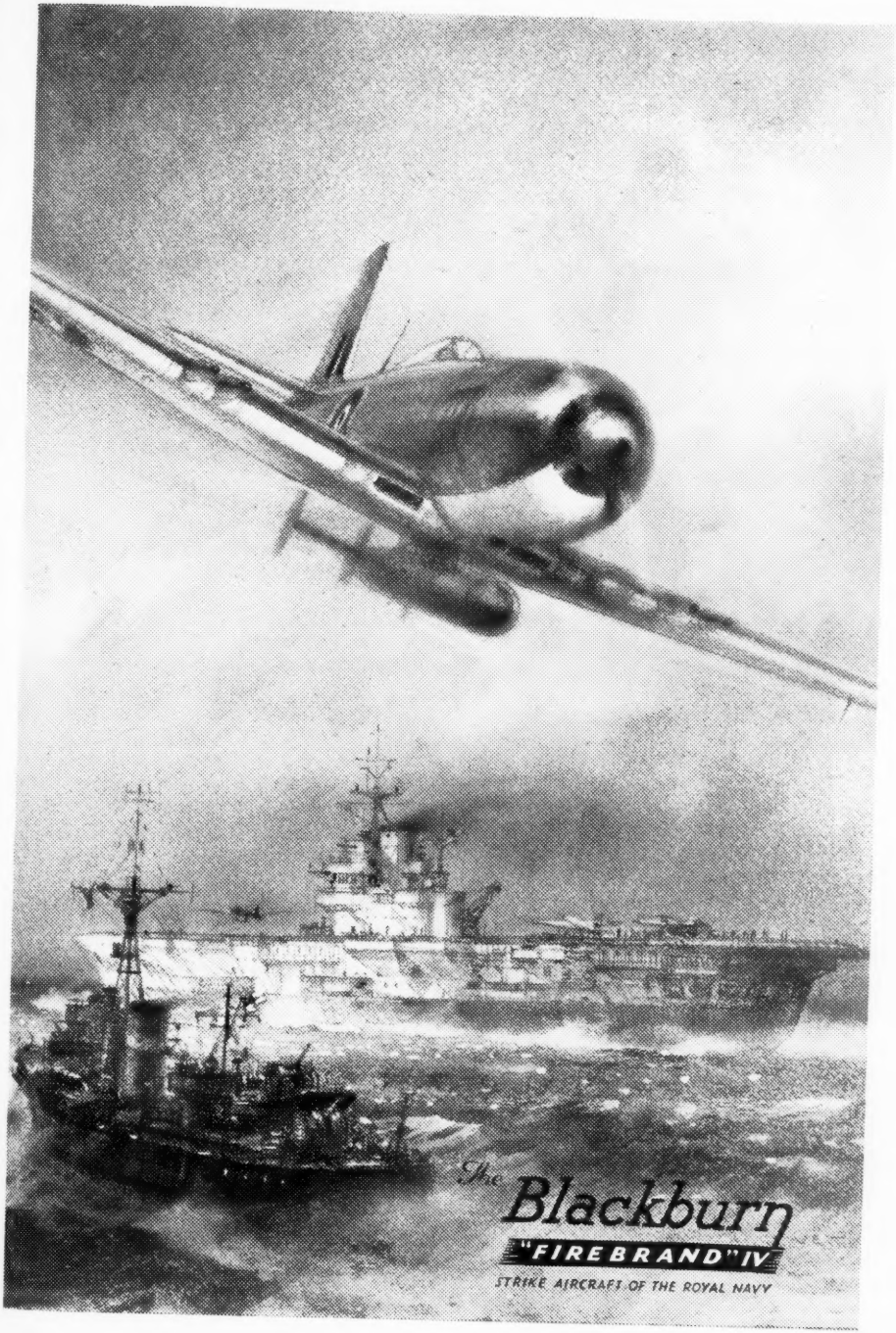
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The Journal of The Royal Aeronautical Society was founded in 1897 in succession to the Annual Reports. For the opinions expressed in Papers that are signed or initialled the authors alone are responsible. None of the papers or paragraphs must be taken as expressing the opinion of the Council of The Royal Aeronautical Society unless such is definitely stated to be the case. **The Journal is published monthly at the Offices of the Society, 4, Hamilton Place, Piccadilly, London, W.1.** Subscriptions per annum, £4 13 0 including postage; Single Numbers 7/6 or 7/9 post free. All communications for publication in the Journal, or on general matters affecting the Society should be addressed to — The Secretary, J. Laurence Pritchard, 4, Hamilton Place, W.1. The Society's Bankers are Messrs. Coutts and Co. All communications respecting Advertisement matters should be addressed to—The Advertisement Manager, Journal of The Royal Aeronautical Society, 4, Hamilton Place, W.1.
Telephone: Grosvenor 3515-3516-3517. Telegraphic Address: Didaskolas, Audley, London.

AIRCRAFT ENGINE OIL COOLING

by

FRANK NIXON, B.Sc. (Hons.), M.S.A.E., Fellow

Mr. Nixon took his degree at Victoria University, Manchester, in 1926. *He served in the Engine Department of the Bristol Aeroplane Co., Ltd., from* 1929 *until he left in* 1942, *as Design Manager, to join Rolls-Royce, Ltd., as Chief Power Plant Quality Engineer. In addition to his usual duties during the past year he has acted as Consultant to the C.R.D. Ministry of Aircraft Production, on engine heat exchange equipment used by the R.A.F. Mr. Nixon was elected an Associate Fellow in* 1931 *and became a Fellow in* 1940.

A MEETING OF THE Royal Aeronautical Society was held in the lecture hall of the Institution of Mechanical Engineers at Storey's Gate, St. James's Park, Westminster, London, S.W.1, on Thursday, the 18th October, 1945, when a paper on "Aircraft Engine Oil Cooling," by Mr. F. Nixon, B.Sc., M.S.A.E., F.R.Ae.S., was presented and discussed.

SIR FREDERICK HANDLEY-PAGE, who presided, said that Mr. Nixon, after serving for some years in the engine department of the Bristol Company, had joined Rolls-Royce, Ltd. During the past twelve months, in addition to his usual duties, he had acted as consultant for C.R.D. on matters relating to heat exchange equipment.

As this was the first occasion on which Air Marshal Sir William Coryton, the new C.R.D., was present at a meeting of the Society, the Chairman offered him a particularly hearty welcome, at the same time thanking him and his predecessors for all that they had done for the Society and looking forward to their help in the future.

MR. NIXON presented his paper.

1. INTRODUCTION.

When at the beginning of the year I was invited by the Council of the Society to give a Paper on oil cooling I had had little more than average contact with the subject, but nevertheless I was startled when I discovered the extent of my ignorance. Inquiry showed that this lack of knowledge was fairly general, and that apart from highly technical articles by specialists, there was remarkably little up-to-date published data. I have accordingly approached the problem primarily from the point of view of the user and operator, and have tried to give a general review of the subject on broad lines.

The oil system has been by no means the most reliable and trouble-free section of aircraft during the war, and it is hoped that

a more widespread understanding of the problem will lead to action which will remove one worry at any rate from the mind of the civil airline operator. By great good fortune I have, since the first invitation to present this Paper, spent some time in Canada and the United States, so that it has been possible to cover recent developments over there, which have succeeded in raising the standard of reliability of the oil cooling system to a far higher level than that which we have enjoyed in this country.

It is proposed to concentrate on the cooling of the lubricating oil of engines installed in aircraft. Dealing first with the need for cooling, and the magnitude of the quantities involved, there follows a review of the methods of cooling adopted in the past. The theory of cooling and of oil temperature regulation is covered sufficiently to convey a working understanding of the problem, and current practice is described. The causes and cure of the troubles which have been experienced are dealt with at some length, and the final section is devoted to the most recent development and probable future trends.

The allied subjects of engine lubrication systems, aircraft oil systems, oil tank design, de-aeration, and oil dilution, with altitude effects and operation at low temperatures, are only brought in where they concern the main topic of cooling. They are worthy of a separate study, which it is hoped will be made before they have been rendered obsolete, or at any rate less important, by the general adoption of turbine power units.

2. THE NEED FOR OIL COOLING.

2.1. *Requirements.*

The temperature of the lubricating oil entering an aircraft engine must be controlled between certain limits for the following reasons:—

i. To maintain the chemical stability of the oil, in order to avoid the formation of sludge, piston ring gumming, and corrosive attack of bearing alloys, and hence to increase the life of the engine between overhauls.

ii. To maintain the temperature of bearing alloys at a figure compatible with adequate strength for their duty.

iii. To keep the viscosity of the oil between reasonable limits, thereby reducing bearing friction and bearing temperature.

iv. By limiting the maximum viscosity of the oil (*i.e.*, its minimum temperature), to reduce friction losses and to ensure that an adequate supply of oil reaches all parts of the engine at all times.

In its passage through the engine the oil picks up a considerable amount of heat (*a*) by conduction from the hot pistons, cylinder walls, and valves; (*b*) by viscous friction in the bearings; and (*c*) by churning in the reduction gear and other transmissions of the engine, and possibly in the crankcase section. Before it leaves the engine, however, the oil gives up some of the heat so gained to the crankcase, oil sump, and gear casings.

2.2. *Amount of Heat Picked up by Oil.*

The net amount of heat which is picked up by the oil varies considerably with the type and the individual characteristics of the engine.

Table I gives figures of oil flow and heat picked up by the oil, for various engines. It is regretted that this data is not more complete, but there is little published information, and some of that reproduced has been difficult to come by. Moreover, the figures for a given type of engine may vary widely (as much as $\pm 30\%$) throughout a production batch. However, an effort has been made to select average figures, and it is felt that they give a fair comparative picture of the lubrication characteristics of the different types.

This tabulation shows that whereas for engines of fifteen years ago the heat-to-oil was of the order of 2% to 3% of the power output of the engine, only in the unique case of the Merlin and Griffon engines is it nowadays less than double this figure, while the average is around 8% and in some engines much higher.

2.3. *Factors Affecting Amount of Heat-to-Oil.*

Reasons for this considerable increase, common to all types of engine, are the close cowling of engines, the general adoption of airscrew reduction gearing, the great increase in the number of accessories driven by the engine, the use of constant speed airscrews operated by the main oil supply, and the greatly increased specific power output.

There are other factors, however, which have had a far greater effect than these, and which are responsible, too, for the wide variation in the heat-to-oil figures.

It can be seen from Table I that the mode of cooling is no longer the predominant factor, nor is it, as many believe, largely a matter of high oil flow. A high rate of oil flow, if properly controlled, is entirely beneficial, although in some engines it does lead to increased heating of the oil. The jacket temperature in liquid-cooled engines was shown by Caminez (Ref. 1) to have comparatively little effect. In the Merlin engine the considerable increases in power have been obtained primarily by raising the boost pressure, and increases in power output produce only small increases in the heat to coolant.

The main variables affecting the amount of heat picked up by the oil are, excluding those already mentioned above:—

(*a*) Engine Power.
(*b*) Crankshaft speed.
(*c*) The extent to which oil is used to cool the pistons.
(*d*) The use of main lubricating oil in the supercharger drive, particularly when this is of the fluid-flywheel type.
(*e*) Cooling air temperature, with air-cooled engines.
(*f*) Amount of churning of oil inside the engine, which is a matter of rate of flow and adequate drainage.
(*g*) The amount of cooling offered by the crankcase.
(*h*) In some engines, the extent to which the cylinder skirts or sleeves protrude into the crankcase.

It is only recently that designers of air-cooled engines have admitted frankly that they use engine oil to cool the pistons. (Refs. 2 and 3.) Nowadays, with hardly an exception, radial engines of high power output have oil jets in the crankshaft, which continuously spray oil directly on to the pistons, or in some cases, on to the connecting rods which in turn splash it on to the pistons. Some go further, by having pistons specially designed, with fins or stalactites on the underside of the piston crown, ribs behind the piston skirts, and a spacious internal layout to allow free circulation of the cooling oil.

The need has not yet arisen on most liquid-cooled engines, despite their extremely high specific outputs. It is an arresting thought to contemplate the boost pressures which might be attained if piston cooling jets were installed in the Merlin engine!

It is interesting to note that, in the case of Bristol engines at least, the adoption of the sleeve valve has had little effect on the heat-to-oil.

Many engines with 2-speed and 2-stage superchargers use main engine oil to operate the clutches. Where the oil merely moves the pistons, there is little heat added to the oil, but where hydraulic transmissions are used, as in recent Pratt and Whitney engines and the Daimler Benz D.B.601A engine, there may be considerable heat added, depending upon the way in which the transmission is used. If it brings in the second stage only at altitude, it is relatively innocuous, but in the D.B.601A engine the fluid flywheel is made to slip when maximum power to blower is not required. At sea level the heat added to the oil by this drive is equivalent to 28 H.P. (Ref. 4.)

It will be noted that fitting a two-stage blower to the Merlin engine has had little effect on the heat-to-oil. In this case the drive is entirely mechanical.

Even with fully-cowled engines, some heat is given up by the oil to the crankcase. This will be influenced by the temperature of the air flowing over the crankcase, in a certain case the temperature rise of the oil passing

TABLE

OIL FLOW THROUGH ENGINE

Engine	*Engine Cooling*	*Power Condition*	*R.P.M.*	*B.H.P.*	*H.P. to Oil*
PRE-WAR TYPES					
Napier Lion XIa	Liquid		2350	530	14
R.R. Condor III	,,		1900	684	14.4
R.R. "F" Type	,,		2250	480	11
R.R. "R" Type, 1929	,,	Schneider Trophy	2900	1900	40
R.R. "R" Type, 1931	,,	,, ,,	3200	2300	116
Bristol Jupiter IV	Air		1575	380	8.4
Bristol Pegasus IIL	,,		2000	600	17.5
P. & W. Wasp Jr.	,,	Take-off	2300	450	17
P. & W. Wasp	,,	,, ,,	2250	600	22.6
CURRENT TYPES					
R.R. Peregrine	Liquid	Max.	3000	885	71
R.R. Merlin 24	,,	Take-off	3000	1640	65
		Climb	2850	1270	55
R.R. Merlin 100	,,	Take-off	3000	1710	65
		Climb	2850	1285	55
R.R. Griffon 64	,,	Take-off	2750	2300	95
Allison V-1710	,,	Take-off	3000	1325	155
Daimler-Benz D.B.601A	,,		2400	1000	86
Napier Sabre IIa	,,	Combat	3700	1875	309
Bristol Mercury 30	Air	Combat	2750	855	48
Bristol Taurus II, VI, VII	,,	Climb	2800	985	63
Bristol Hercules VI†	,,	,,	2400	1356	86
Bristol Hercules 100†	,,	,,	2400	1515	97
Bristol Centaurus IV†	,,	,,	2400	2095	152
P. & W. Twin Wasp†	,,	,,	2550	1050	66
,, ,, ,,	,,	,,	2550	1100	71
P. & W. Double Wasp†	,,	Take-off	2700	2000	177
		Climb	2550	1675	136
P. & W. Wasp Major†	,,	Climb	2550	2500	284
W.A.C. Cyclone R.1820†	,,	Take-off	2500	1200	71
W.A.C. R.2600†	,,	Climb	2400	1500	90
W.A.C. R.3350†	,,	Climb	2400	2000	132

NOTES † Engine embodies Piston Cooling Oil Jets.

* Temperature Rise Calculated from Heat-to-Oil Figures.

‡ Supercharger Transmission gives 28 H.P. to Oil at Take-off. Reference 4.

I

AND HEAT PICKED UP BY OIL

Oil Flow G.P.H.	*H.P. Oil B.H.P. %*	*Oil Flow H.P. Oil*	*Normal Max. Oil Temps at Engine*			*Remarks*
			Out	*In*	*Rise*	
285	2.64	20.4	94	77	17	
460	2.11	32.0	97	88	9	
200	2.29	18.2	74	60	14	The Prototype of the Peregrine
380	2.10	9.5	113	80	33	
665	5.06	5.7	120	65	55	The Prototype of the Griffon
51.5	2.21	6.1	90	55	35	
100	2.82	5.7	120	80	40	
346	3.78	20.4	100*	85	15.4*	
406	3.77	18.0	102*	85	17.4*	
594	8 03	8.4	127	90	37	
780	3.96	12.0	117	90	27	2-Speed Blower
750	4.33	13.6	113	90	23	
780	3.80	12.0	117	90	27	2-Speed 2-Stage Blower
750	4.28	13.6	113	90	23	
1150	4.13	12.1	116	90	26	2-Speed 2-Stage Blower
1040	11.70	6.7	132*	85	47*	
640	8.60	7.5	117	75	42*	Hydraulic Blower Drive‡
2400	16.50	7.8	130*	90	40*	2-Speed Blower
355	5.62	7.4	144	100	44	,, ,,
375	6.40	6.0	125	70	55	,, ,,
495	6.35	5.7	146	90	56	,, ,,
505	6.40	5.2	142	80	62	,, ,,
700	7.26	4.6	150	80	70	,, ,,
634	6.28	9.6	118*	85	33*	Single-Speed Blower
655	6.45	9.2	119*	85	34*	2-Speed Blower
1165	8.85	6.6	133*	85	48*	2-Speed 2-Stage Blower
1100	8.12	8.1	124*	85	39*	
2130	11.36	7.5	127*	85	42*	2-Speed 2-Stage Blower
666	5.91	9.4	118*	85	33*	
800	6.00	8.9	120*	85	35*	2-Speed Blower
1000	6.60	7.6	127*	85	42*	2-Speed Blower

through the engine increasing 1° C. for each 3° C. rise in cooling air temperature.

With liquid-cooled engines sump design may be most important. Fitting a deeper sump to the 1931 Schneider Trophy Rolls-Royce "R" engine reduced the oil outlet temperature by 20° C. (Ref. 5.)

While a high rate of flow, as is the case with the Merlin engine, is not necessarily conducive to a high heat pick-up, nevertheless in some engines it is found essential to keep the flow to as low a figure as possible, for a variety of reasons. Since a low rate of flow presents the oil cooler designer with a most difficult problem, it is interesting to review the factors involved.

Table I shows a wide variation in rate of oil flow with different designs of engine. This variation has existed for a long time, and it is bound up with a large number of somewhat indeterminate factors, not the least being the personal predilections of the designer, and the traditional design of crankshaft, connecting-rod big end bearing, etc., of a given type of engine. In some cases, quite fortuitously, the draining and scavenging of oil from the crankcase has presented no difficulties. In others, nothing short of complete redesign could improve matters, and there has in consequence been a tendency to keep the oil flow to a minimum.

It is interesting to find the late Group Captain G. B. Hynes quoting, twenty years ago (Ref. 6) an oil flow of 18 g.p.h. in one engine and 450 g.p.h. in an engine of half the power output. Four years later, Mr. Foord (Ref. 7) instanced flows of 60 g.p.h. and 400 g.p.h. in two engines of the same power output.

At that time there was expressed a desire for a study of the factors involved, but little has been published, save for one valuable paper (Ref. 2) dealing specifically with the Wright Cyclone engine. This shows that the rate of oil flow depends upon many factors, and describes how a crankshaft oil jet was fitted in the first instance in order to allow the flow of oil to be regulated. The adoption of big end bearing seals, following Bristol practice, led to an improvement. However, a wide variation in oil flow in a particular design of engine may still occur, especially where a built-up big end assembly precludes the effective sealing of the bearing.

2.4. *Cold Starting.*

The importance of adequate lubrication during the first moments of an engine's running must not be overlooked. It takes an appreciable time for oil to reach the parts more remote from the pressure pump, and it is essential therefore that the oil shall quickly attain a temperature permitting this. This is primarily a matter of engine lubrication system design, which may embody gallery pipes in the case of in-line engines, or devices permitting of extremely high starting pressures, as in the Bristol High Initial Oil Pressure scheme first introduced many years ago, or in thermostatically-controlled valves as in the latest American radial engines. Partial circulation arrangements in the oil tank are also of considerable assistance, as, too, is the dilution of the oil with fuel, which is now used universally in America, but which has been all too reluctantly adopted in British installations, chiefly because the need for it does not normally arise during home operations.

In addition, the control of the oil cooler, which will be dealt with later, has an important effect.

2.5. *Turbine Engines.*

In jet propulsion and I/C turbine units, with the complete absence of reciprocating parts, there is a marked reduction in the heat picked up by the oil. In the engines so far developed, the main factors have been the type of journal bearings used, and the presence or absence of airscrew reduction gearing. In one quite powerful jet propulsion unit, having anti-friction journal bearings, there is no need at all for external oil cooling. Another, having plain journal bearings, has a small oil cooler dissipating 14 H.P., which is under 1% of the equivalent engine horsepower at normal speeds of flight and, of

course, less at the higher aircraft speeds possible with jet engines. Turbine engines driving airscrews through reduction gearing have a higher oil heat pick-up, but even here the heat horse power is only of the order of 2% of the airscrew shaft horse power.

3. THE DEVELOPMENT OF OIL COOLING

3.1. *Introduction.*

It was a long time before the desirability, and later the need, for an oil cooler as a separate item of equipment gained general acceptance. Up to the end of the first World War the power output of aircraft engines seldom exceeded 300 H.P. and the dissipation of the 4 or 5 horse power picked up by the oil presented no problem. The engines and oil tanks were usually completely exposed,

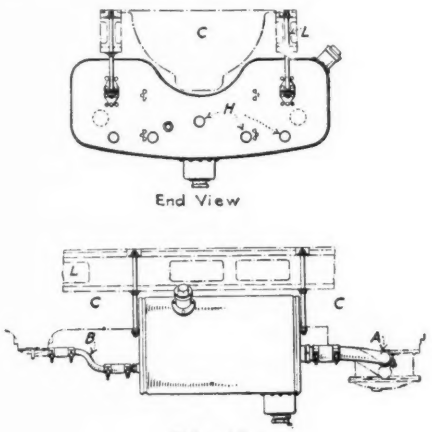


Fig. 1

Sketch showing wet sump of early aircraft engine with air tubes for oil cooling (Marks: " The Airplane Engine ").

and heat losses in the pipe lines and oil tanks, together with the cooling of the exposed crankcase, were sufficient to maintain the oil temperature at an acceptable level.

3.2. *Early Oil Coolers.*

A few designers did appreciate the need of oil cooling in their individual engines. One of the earliest marks of Rolls-Royce Eagle engine had a finned sump, with cooling air tubes running through it, but this was discontinued in later models. Finning of the sump has, incidentally, persisted to the present day in the case of the Gipsy and some other engines. An early example of a sump tank with cooling air tubes is shown in Fig. 1 (Ref. 8).

During the war, according to Mr. C. Anderton Brown (Ref. 9), honeycomb oil coolers were fitted to the engines in Zeppelins, but perhaps the first example of an oil cooler, fitted as a separate ancillary to an aircraft engine, is that shown in Fig. 2 (Ref. 10). This is the American Thomas-Morse engine of 150 H.P. in 1917, with a rudimentary oil cooler comprising a coiled pipe fitted between the oil pressure pump and the front journal bearing of the crankshaft, where the oil entered the engine.

Most of the engines at this time had wet sumps, until aerobatics dictated otherwise, and for several years afterwards, additional cooling of the oil was achieved by finning the oil tank, or by fitting air tubes to it as in the tank shown in Fig. 1.

It would appear that during the early 1920's, with increasing power outputs, oil temperatures began to rise, and, as usual, the engine designer put the blame on to the aircraft man. In 1925 we find Sir Roy Fedden (Ref. 11) complaining bitterly about the haphazard installation of oil systems, and in self-defence offering what must be one of the first British oil coolers. This was quite a neat tubular design, "which should be slung about 2" clear of the fuselage." Shown in Fig. 3, this was less than four inches in diameter, and only weighed 3 lb. empty.

Previous to this, efforts had been made to extract heat from the oil, and at the same time to assist in vapourizing the fuel, by passing the return oil through a jacket on the induction manifold. Hall-Scott had an American patent on this before 1919, Bristol jacketed the air intake as shown in Fig. 4 about 1925, and Pratt and Whitney had a similar scheme in 1931.

Papers by Mr. Fedden in 1925 (Ref. 11) and Mr. Foord in 1929 (Ref. 7) gave rise to discussions on oil cooling which are most interesting, as many of the participants are still actively engaged in aviation, and the views which they then expressed hold as good to-day in the majority of cases. It has been our misfortune that the leads they gave were not followed up either then or subsequently.

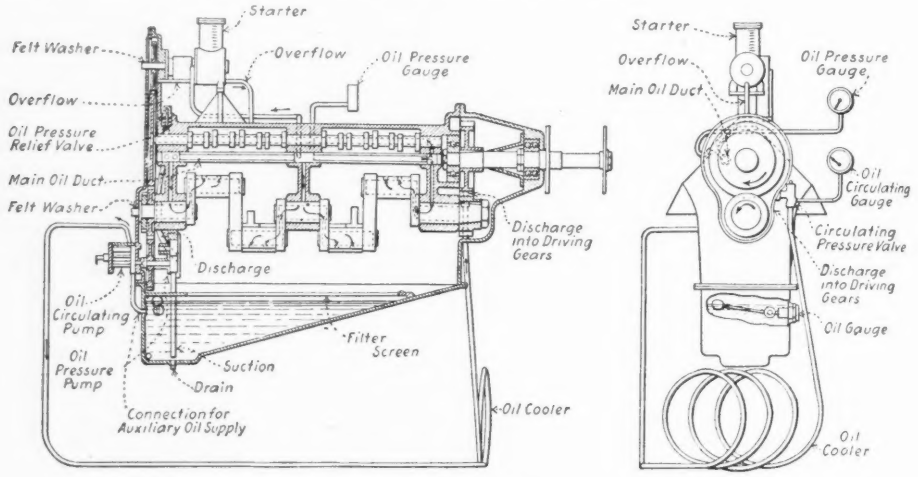


Fig. 2
Rudimentary Oil Cooler on Thomas-Morse Aircraft Engine (Ca. 1917).
(Page: " Aviation Engines.")

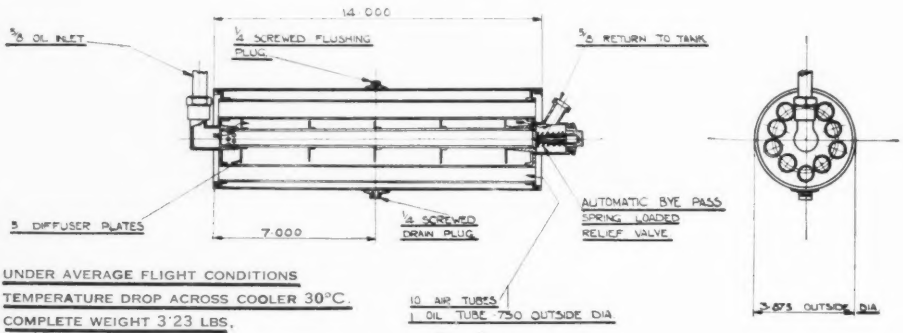


Fig. 3
Bristol Tubular Oil Cooler.

About this time much work was being carried out at the R.A.E., and this was given a stimulus by the suddenly increased oil cooling demands of the Schneider Trophy engines. This work is described in R. & M. 1486, "Oil Cooling for Aircraft," by Major B. C. Carter, a report which gives descriptions of many of the types of oil cooler tried up to that time. It is worthy of note that these three Papers, written from 15 to 20 years ago, were the last ones to deal with oil cooling from the engine designer's viewpoint.

Meanwhile, about 1927, the Wright Company of America, which was then producing liquid-cooled engines, including the Tornado T-3 of 650 H.P., developed the oil temperature control system in Fig. 5 (Ref. 12). This is extremely interesting as employing water-cooling for the oil, with a honeycomb matrix

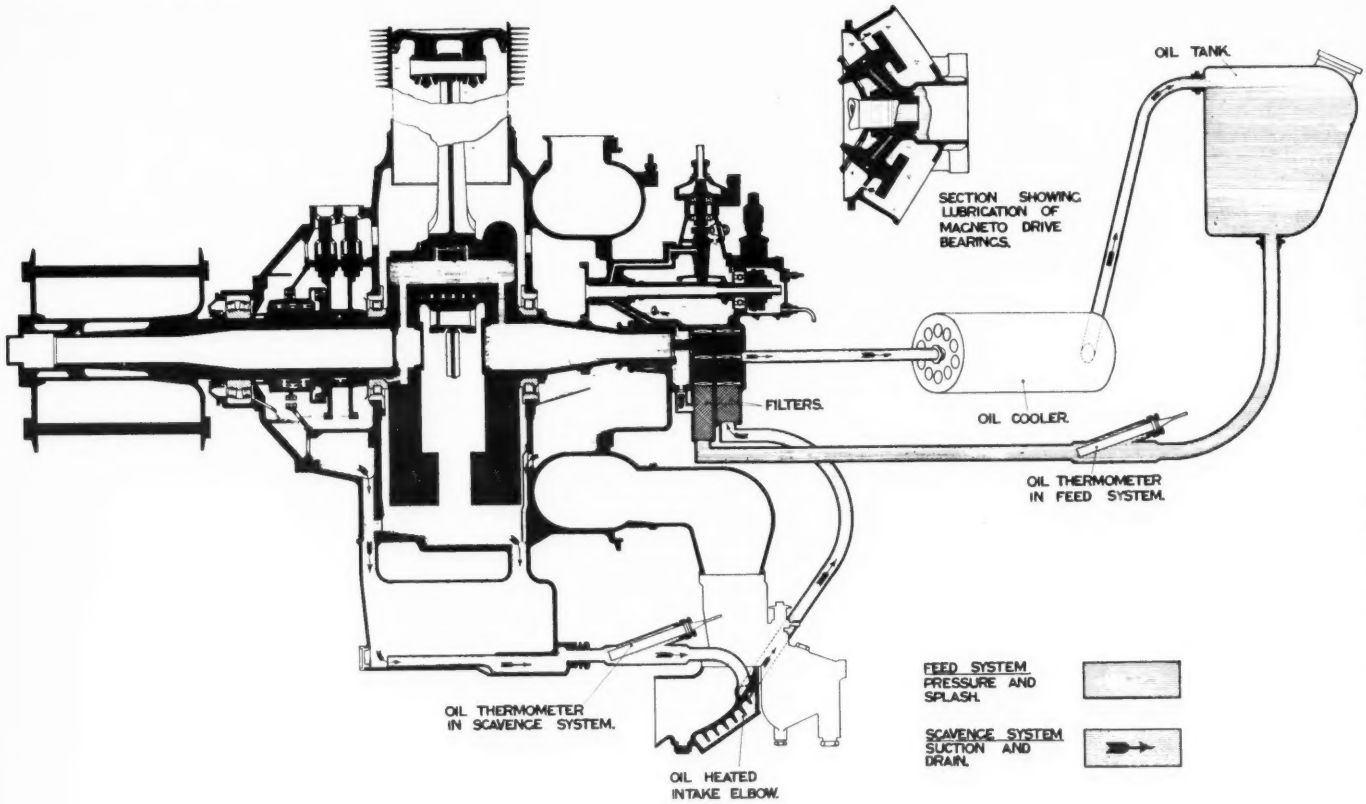

OIL TANK.
SECTION SHOWING LUBRICATION OF MAGNETO DRIVE BEARINGS.
FILTERS.
OIL COOLER.
OIL THERMOMETER IN FEED SYSTEM.
OIL THERMOMETER IN SCAVENGE SYSTEM.
OIL HEATED INTAKE ELBOW.
FEED SYSTEM PRESSURE AND SPLASH.
SCAVENGE SYSTEM SUCTION AND DRAIN.

Fig. 4

Oil system of Bristol Jupiter VI Engine, showing oil-heated air intake and tubular oil cooler (1926).

(Fig. 6, Ref. 12), as used in radiators. This form of oil cooling, which will be described later, has during the past year or so been revived in the latest North American Mustang.

In Britain it is understood (Ref. 9) that certain aircraft designers had tried to use honeycomb radiators for cooling oil, but had found that with the higher pressures and temperatures they were constructionally inadequate. At about this time there were, in addition to the Bristol tubular oil cooler, the

During all this time there had been a marked reluctance on the part of aircraft designers to install oil coolers, and something of a feeling of shame on the part of the engine designers that their engines should be so badly designed as to increase the temperature of the oil. In 1929 some installations could still function without oil coolers (Ref. 14), but by 1931 cooling of the oil "by placing in the slipstream a small efficient radiator" (Ref. 15) was generally accepted.

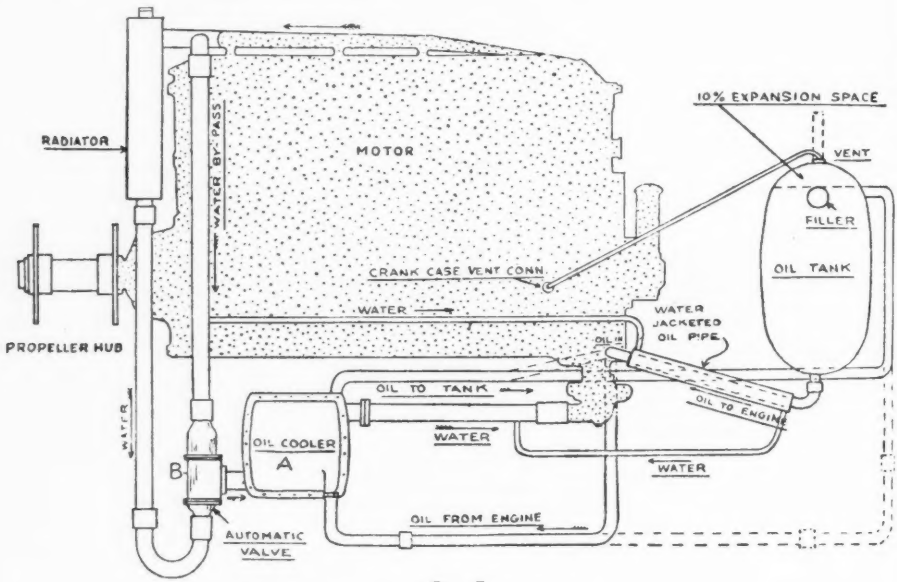


Fig. 5
Oil system of Wright Tornado Engine, with water-cooled oil cooler (Ca. 1927).
(Page: "Modern Aircraft.")

trombone type, Fig. 7 (Ref. 13), which was developed by Mr. Anderton Brown, and the Vickers-Potts shown in Fig. 8. There were other efforts, such as the centrifugal cleaner-cooler, and plate oil coolers working on a similar principle to the Vickers-Potts, but this last, which was introduced in 1928, was most successful, and was fitted almost exclusively to aircraft of the day. Its demise came when fully-cowled engines and the increasing speeds of aircraft led to the suppression of excrescences, although it survived for a while even with fully-cowled engine installations.

The Schneider Trophy contests of 1929 and 1931 did a great deal to further the knowledge of oil cooling. The heat-to-oil suddenly rose to 40 H.P., and then to 70 H.P., while aircraft speeds increased to 329 m.p.h., and then to 409 m.p.h. Drag had become vitally important, and for the special purpose of the contests it was possible to resort to cooling means impracticable for military or transport aircraft. The final solution consisted of surface coolers along the sides and bottom of the fuselage, with a spray cooler in the tail fin, utilizing the metal skin of the fin as the cooling

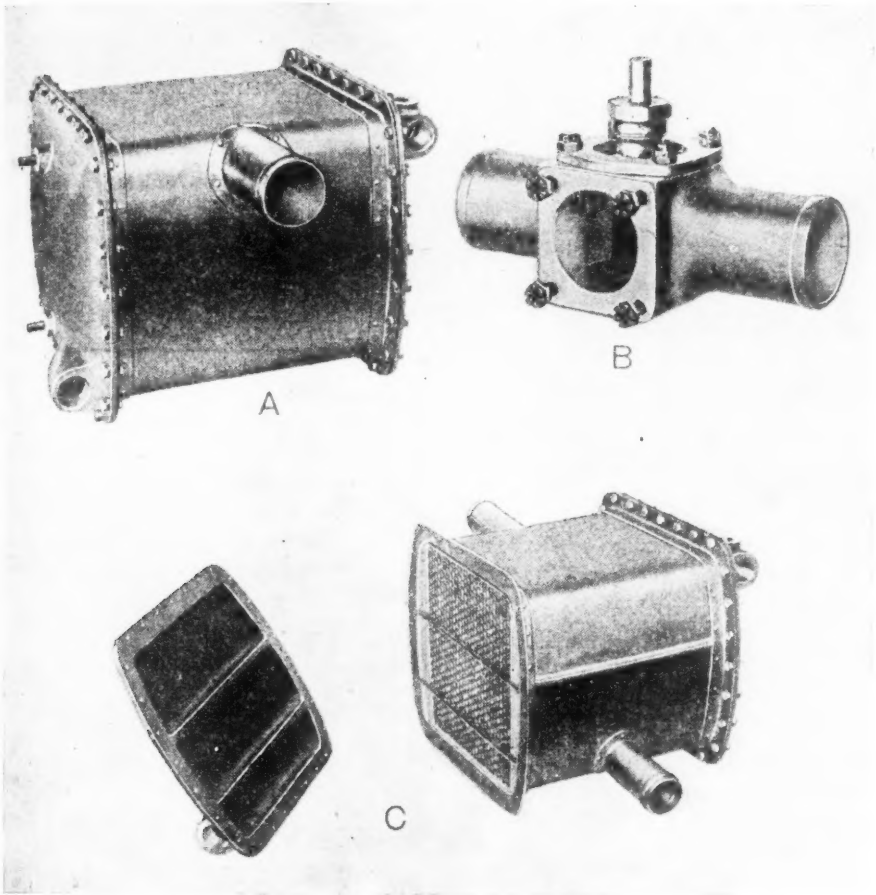


Fig. 6
Parts of Wright Oil Temperature Control System.
A—Heat Transfer Unit or Oil Cooler.
B—Three Way Valve.
C—Heat Transfer Unit with End Plate Removed.
Water-Cooled Oil Cooler used with Wright Tornado Engine (Ca. 1927).
(Page: " Modern Aircraft.")

surface. This combination gave a temperature drop of 60° C. (Ref. 5) between ingoing and outlet oil, a figure until then never reached.

In 1933 Bristol produced an annular oil cooler for use with a Townend ring (Fig. 9). Although this type did not persist, it was revived in modified form by the Germans, in the F.W.190 machine.

About this time there was produced the Robertson aluminium tubular oil cooler shown in Fig. 10. This was notable as pioneering the application of mechanical tube joints, using thin aluminium tubes. It was not economical in air consumption, however, and it too died as the need for ducting the heat exchanger became appreciated.

The first honeycomb type oil coolers appear

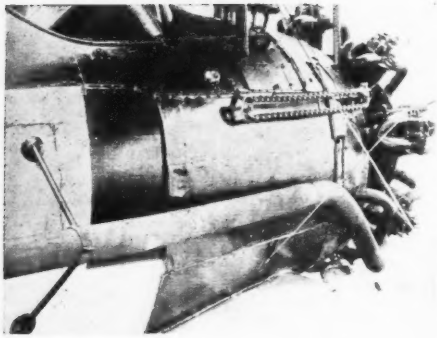


Fig. 7
Trombone Type of Oil Cooler.

to have been evolved in the United States, only a short time before the R.A.E. type of cross-flow series honeycomb oil cooler was developed, independently, in this country. The first British application was about 1934, in the Douglas installation of the Pegasus II engines in the D.C.2 aircraft. Fig. 11 shows a rear view of this installation, with a 9″ diameter United Aircraft Products oil cooler mounted in a duct, on the firewall.

3.3. *Introduction of Current Types.*

In 1935 the late Captain Barnwell made an effort to cool the oil of the Bristol Mercury engines in the "Britain First" aircraft (the forerunner of the Blenheim) by spraying it inside the leading edge of the wings. This proved to be complicated and costly, however, and 3-element R.A.E. drum type oil coolers were adopted.

During the same year another notable step forward was announced, by Mr. A. G. Elliott (Ref. 16). This covered ducting of radiators and oil coolers for Rolls-Royce engines, with control of the airflow by means of duct flaps, and a reduction in cooling drag to 3% of the engine power. The honeycomb oil cooler was mentioned as already being in existence, along with the Vickers-Potts, while the combination of oil cooler and radiator in one block, in a common duct, as so successfully used in the Merlin Lancaster power plants and other Merlin installations, was presaged. At the same time he announced the secondary surface coolant radiator. This has since been adopted for oil cooling also, by Napier and Rolls-Royce.

Historically interesting is the plate-type oil cooler, installed edge-on in the duct ahead of the radiator in the Kestrel Hart aircraft, illustrated in Fig. 12 (Ref. 16).

The years immediately before the war led to the increasing adoption of the honeycomb oil cooler, until it became by far the most important type in use throughout the world. Its main advantages are its small size for a given heat transfer, and its relatively low air resistance, enabling it to be fitted in a duct and permitting of a low drag installation. As an example, a cylindrical oil cooler 16″ diameter and 12″ long is capable of dissipating about 150 H.P. for an air pressure drop of 5.5 inches of water, under temperate ground level conditions, representing a cooling drag of about 3 H.P. assuming no heat recovery.

3.4. *The Honeycomb Oil Cooler.*

Typical honeycomb oil coolers which have been in general use during this war are shown in Figs. 13, 14, 15, 16 and 17. The most common shape, employed almost without exception on air-cooled engines, and almost exclusively in all American installations, is cylindrical (Fig. 13). This shape is the strongest for its weight, and the cheapest to produce, but it is not always the easiest to instal compactly, although an annular variant (Fig. 14) as used in the Hawker Typhoon, had its advantages.

To achieve compactness of installation, some American aircraft designers have installed two circular oil coolers side by side, and an obvious step from this has been to combine the two in one oval-shaped unit, which has recently been standardized by the Army and Navy authorities. It should be emphasized here that American oil cooler shapes are standardized in these two forms, circular and oval, over a range of sizes which covers all existing needs. The advantages to

the enormous production programme are obvious.

In Britain the needs of the aircraft installation have taken priority, and the designer has been reluctant to leave sufficient space for the oil cooler, which in some cases has been smaller than was in fact necessary. This emphasis on spatial economy has led, too, to a variety of shapes. Fig. 15 shows the oil cooler and one radiator element of the Spitfire IX aircraft, and the Mosquito follows generally similar lines. The Merlin Lancaster oil cooler and radiator are shown in Fig. 16, while Fig. 17 shows the oil cooler, intercooler radiator, and main radiator, on a 2-stage Merlin-engined power plant for the Lincoln and Tudor aircraft.

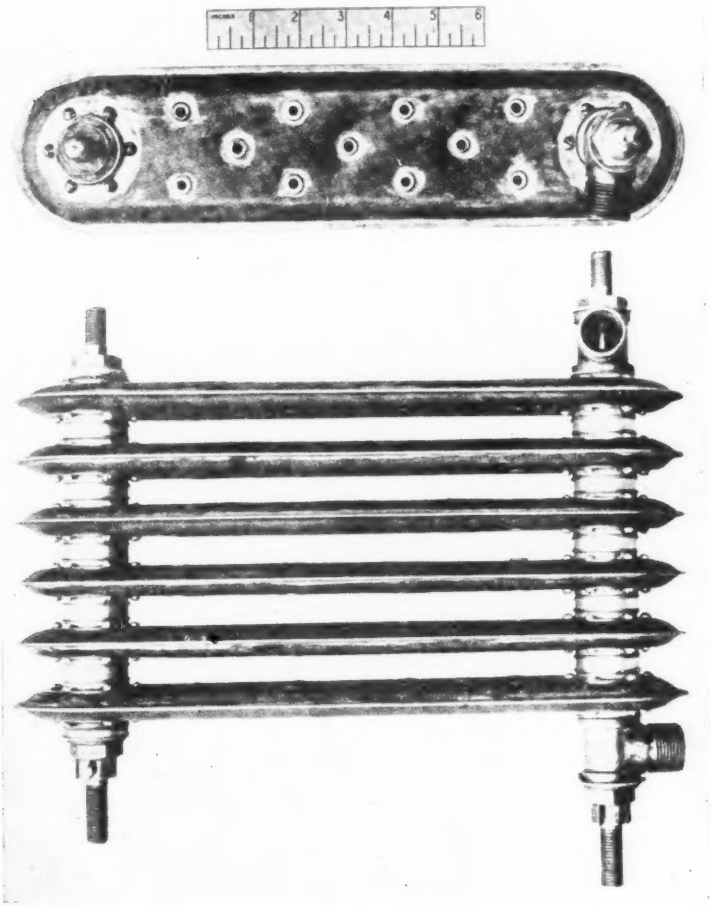


Fig. 8
Vickers-Potts Oil Cooler (from 1928).

With such a variety of designs, and with three lengths of tube standardized, it is difficult to establish standards of comparison for

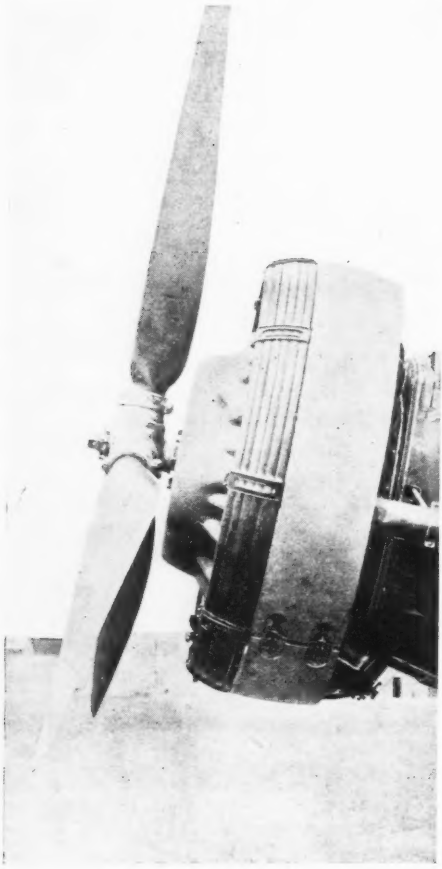


Fig. 9
Experimental Bristol Annular Oil Cooler (Ca. 1933).

oil coolers, more particularly as the specific performances of British and American coolers are based on widely different conditions. Speaking very generally, however, a honeycomb oil cooler of one square foot frontal area is capable of dissipating a maximum of about 130 H.P. in heat-to-oil, for a cooling drag of about 2.5 H.P. and a weight when filled with oil of about 80 lb.

3.5. *German Oil Coolers.*

In Germany during the war there appears to have been less finality of design than in America or this country. To some extent this has been due to the strategic scarcity of copper, which has led to the more or less compulsory adoption of aluminium, though without having obtained much in the way of weight reduction.

Not having made a close study of German types, it is impossible to be definite, but in view of their relatively inferior performances compared with British and American oil coolers on a weight and drag basis, the fact that we find honeycomb tube sizes varying from 7 mm. to 4 mm. may be an indication of efforts to improve matters.

Generally, German oil coolers are characterized by being designed to suit each individual installation. Honeycomb oil coolers with copper matrices are usually rectangular, and much longer and narrower than ours, or curved to suit the cowling outline (Ref. 17).

Two interesting aluminium oil coolers may be mentioned. One of these, used in the Do.18 machine with the Ju.205C engine (Ref. 17) is not unlike the Robertson, save that the tubes through which the oil flows are welded into the header plate. The other, of annular form, installed in the nose of the F.W.190 machine with the B.M.W.801 engine, consists of 60 curved tubes, .320″ bore and .060″ thick, with fins .120″ deep machined out of the solid. These tubes are fitted 5 rows wide and 12 deep into header tanks, by means of gland nuts and olives. This cooler is not light, and its specific heat dissipation compares poorly with contemporary British oil coolers.

4. OIL TEMPERATURE REGULATION.

4.1. *Introduction.*

At this stage, let us consider the problem of heat transfer in an oil cooler and, more particularly, the regulation of the temperature of the oil returned to the tank.

For our purpose, although other designs have their own particular characteristics, a study of the honeycomb oil cooler will cover the main principles involved.

4.2. *Functioning of the Honeycomb Oil Cooler*

Fig. 18 shows a skeleton view of a cylindrical oil cooler. The matrix consists of a bundle of circular copper tubes, about .006″ thick, and usually 9″ or 12″ long, expanded to hexagon section for half an inch at each end, nested together, and sealed by dipping in soft solder. There are two standard tube diameters, $5\frac{1}{4}$ mm. (.210″ American), and .268″ (American). These hexagon sizes are such that the spaces between the tubes are .040″ and .055″ respectively.

The tube nests are suitably encased so that air passes through the tubes, and oil flows over their outer surfaces. Two modes of oil flow may be employed, "cross flow," in which the flow of oil is normal to the tube axes, and "axial flow," in which it is parallel to the axes. With axial flow, when the oil and the air are moving in the same direction, we have "parallel flow," and when they are moving in opposite directions we have "counterflow."

The oil cooler illustrated in Fig. 18 has axial flow, and it is divided up by baffles into a number of passes. The size of each pass is determined by the oil flow and the desired velocity of flow through the cooler, the number of passes and hence the total size of the cooler being fixed by the total heat dissipation required. In the majority of oil coolers, as in this case, the matrix is surrounded by an oil jacket. Oil entering the cooler does so by way of this jacket. When the oil is cold the pressure drop through the matrix will be high and the relief valve will lift, allowing the oil to pass right round the jacket and back to the

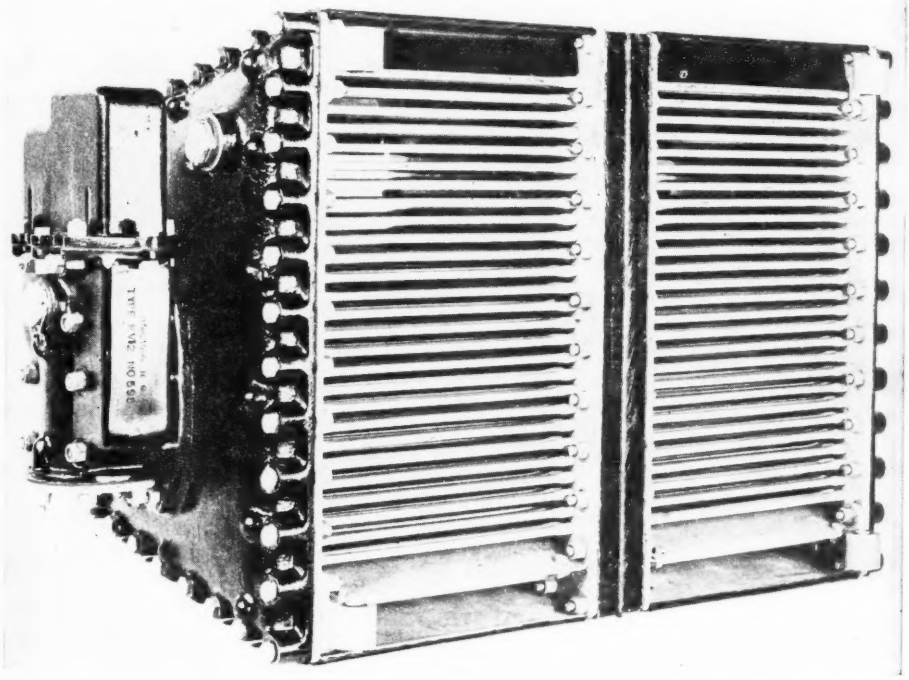
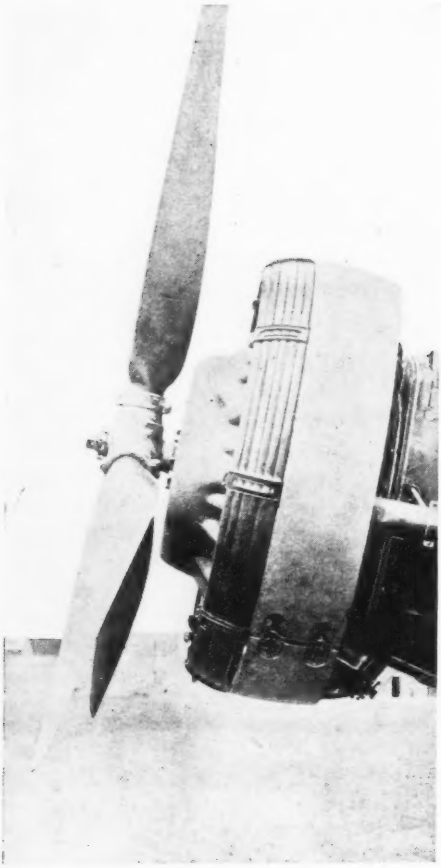


Fig. 10
Robertson Aluminium Oil Cooler, with mechanically-expanded joints.



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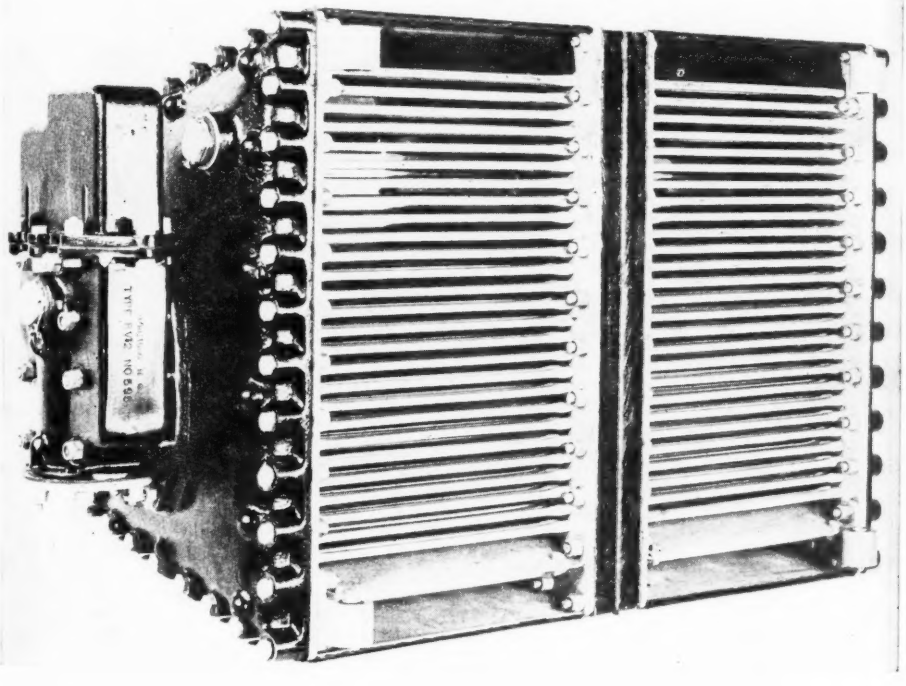

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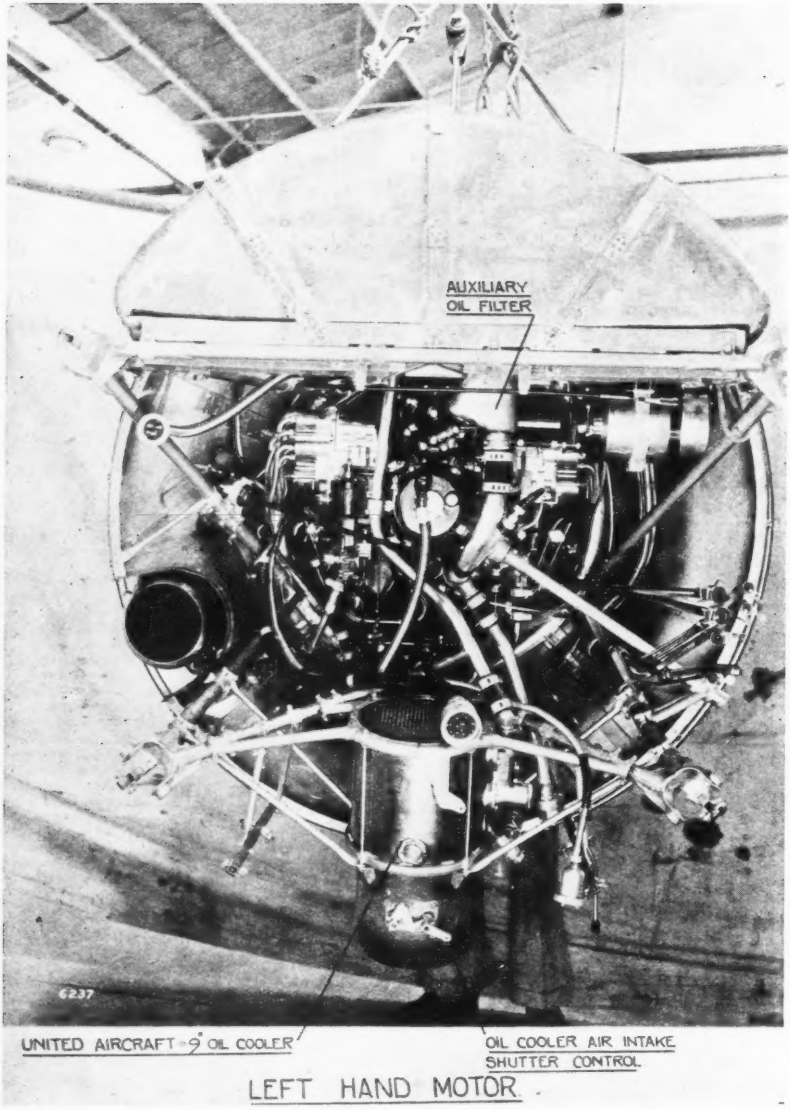


Fig. 11
Early Application of Honeycomb Oil Cooler. UAP. Cooler on Pegasus II—Douglas D.C.2. Power Unit (Ca. 1934).

tank. As the engine heats the oil, the combined effect of the hot oil in the jacket and continually applied oil pressure on the matrix will cause flow through the matrix to begin and the relief valve will close. Hot oil will then enter the bottom pass and successively upwards through the other passes and back to the tank.

4.3. *Heat Transfer.*

The cooling of lubricating oil presents a very different problem from that of cooling water or engine coolant. The rate of flow is relatively small, and the required temperature drop through the cooler is higher, while there are important differences in the physical properties, as shown below.—

Property.	Water.	Oil (D.E.D. 2472B)	Ratio Oil/Water.
Viscosity at—			
0°C. lb./ft. sec.	0.00120	3.98	3320
60°C. lb./ft. sec.	0.000315	0.050	159
100°C. lb./ft. sec.	0.000191	0.011	57.6
Specific Heat at—			
60°C.	1.009	0.53	0.525
100°C.	1.007	0.60	0.595
Thermal conductivity—			
B.T.U./hr./sq. ft./°F./ft. approx. at 30°C. ..	0.356	0.080	0.225

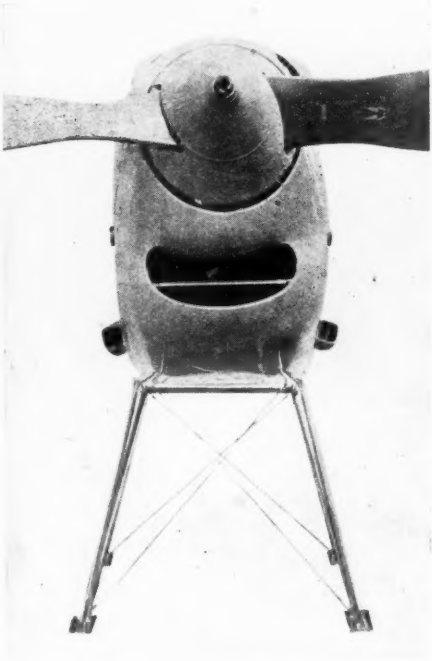


Fig. 12
Plate-type Oil Cooler in Kestrel-Hart.

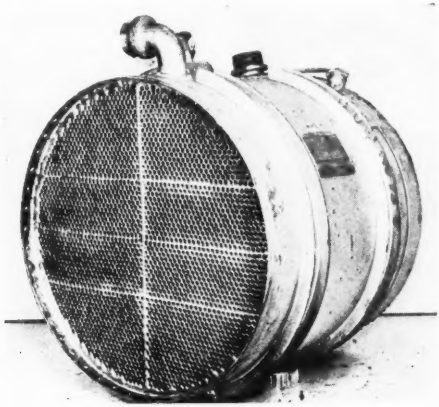


Fig. 13
Typical British Honeycomb Oil Cooler for air-cooled engines.

These are only some of the factors which inspired Mr. Pye to say, in 1929 (Ref. 14), that "oils were devilish tricky things."

There is a widely held fallacy which has persisted for at least twenty years and which it is important to expose. It has been thought by many that the optimum condition for oil cooling exists when there is a minimum oil

flow through the engine with a high temperature rise. In point of fact, as will be shown, the last thing which the oil in the cooler should be allowed to do is to trickle over the cooling surface so as to be allowed a lot of time to give up its heat.

In order to keep the pressure drop within bounds the rate of oil flow (about 1 ft./sec.) through an oil cooler is such that the flow is streamline. There is a stagnant boundary layer of oil in contact with the tube walls, and the thickness of this layer increases with viscosity and decreases with speed. The rate of heat transference is proportional to the temperature gradient across this layer, and inversely proportional to its thickness.

With oil, where the viscosity varies so greatly with temperature (see Fig. 19), it is to be expected therefore that the rate of heat dissipation will vary widely according to the operating conditions of oil temperature, air temperature, air flow, and oil flow.

4.4. *Behaviour of Oil Coolers.*

Setting out the main variables affecting the performance of oil coolers, these are:—

1. Heat to be dissipated.
2. Rate of oil flow through cooler.
3. Temperature of oil entering cooler.
4. Mass air flow through cooler.
5. Temperature of air entering cooler.

In any condition of flight and engine operation, the heat to be dissipated, the rate of oil flow, the air temperature, and the air velocity at the entrance to the cooler will be determined for us. The first two are characteristics of the engine design, while the air conditions depend upon the altitude, the climate, and the speed of the aircraft. The engine designer

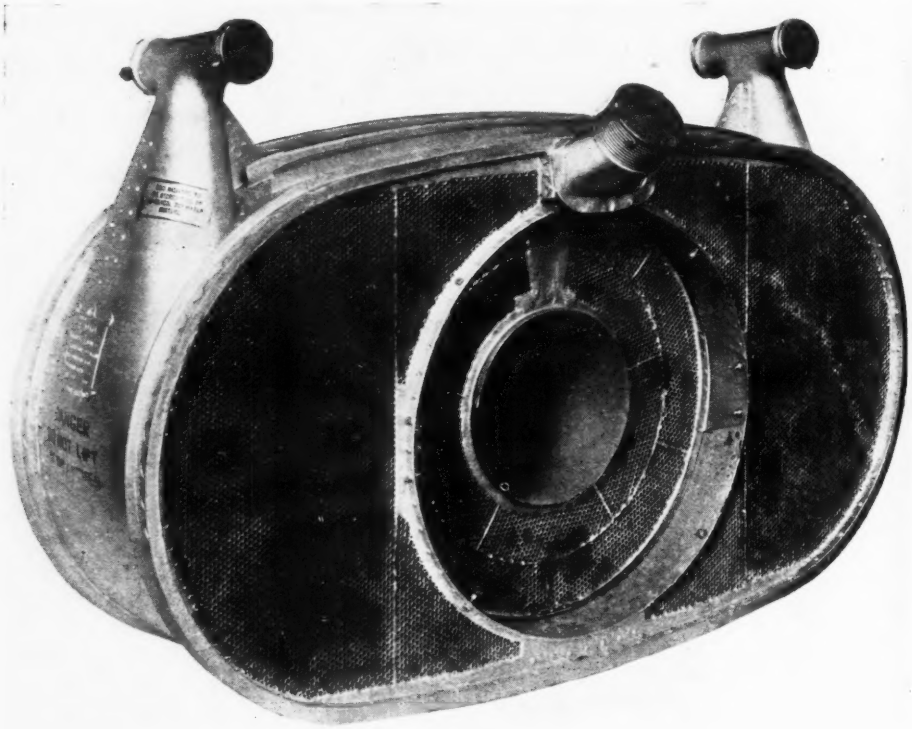


Fig. 14
Annular Oil Cooler and Radiator for Sabre-Typhoon.

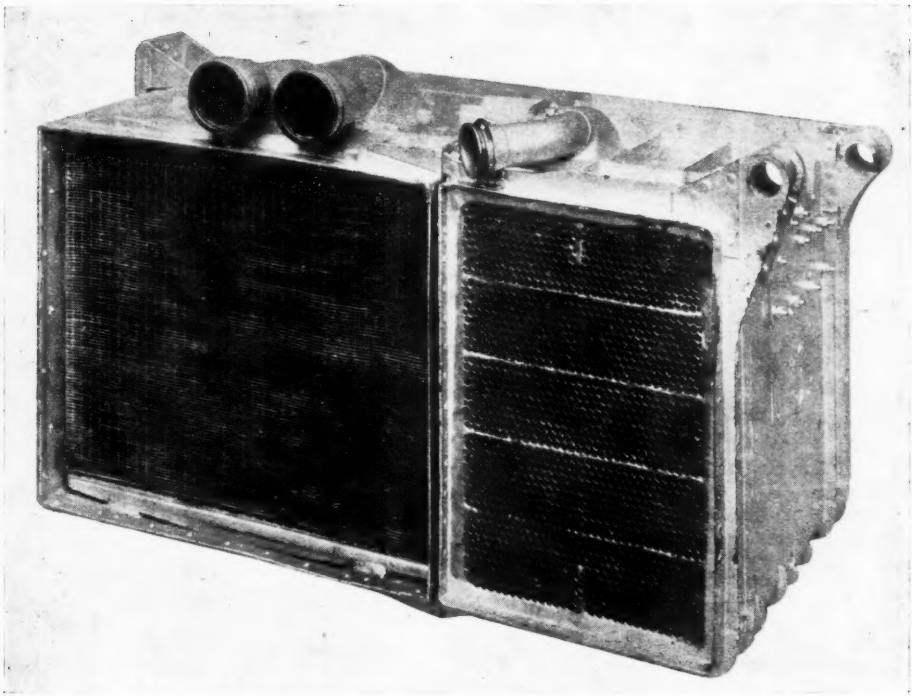


Fig. 15
Rectangular Oil Cooler and Radiator for Merlin-Spitfire IX

will have stipulated the maximum permissible oil inlet and outlet temperatures at the engine, but the actual temperatures achieved, within these limits, will be determined by the oil cooler.

It is important to remember always that a balance must exist between the amount of heat picked up by the oil in its passage through the engine, and the heat dissipated by the cooler. The latter is the product of mass oil flow, the temperature drop through the cooler, and the specific heat of the oil. The inlet and outlet temperatures at the cooler are therefore adjusted by the cooler until it is dissipating the required amount of heat at the air conditions obtaining.

The remaining variable of mass air flow is, in most modern installations, controlled by shutters or a flap at the exit of the duct. This will be discussed later.

In Fig. 20 are given a series of curves based on those produced in a valuable paper by Allfrey (Ref. 18), which deduces mathematically the relationship between pressure drop and the different variables. These curves show: (1) the variation of oil pressure drop with oil flow, for different air temperatures, and constant cooling; (2) the variation of oil pressure drop through the cooler for different oil inlet temperatures and constant cooling; and (3) the pressure drop with different heat dissipations, with constant air and oil-inlet temperatures.

Each set of curves shows the same characteristic, namely, a rapid increase in pressure as the oil flow is reduced, reaching a peak, and afterwards falling to zero when flow has ceased. The rate of increase of pressure, and the peak value, increase as air temperature falls, as oil inlet temperature is decreased, and as heat dissipation increases.

The disadvantages of an excessively low

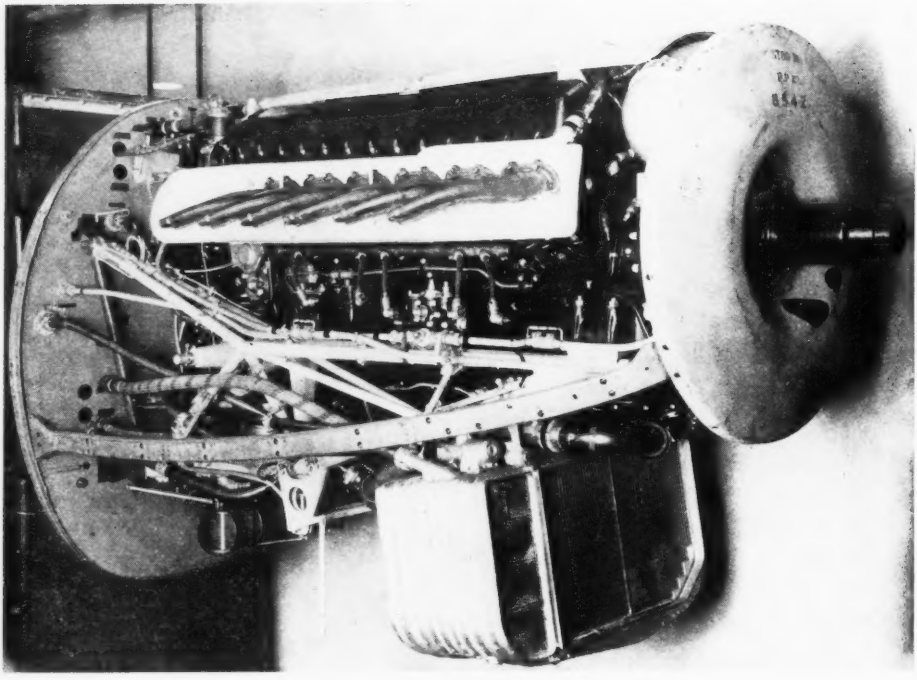


Fig. 16
Oil Cooler and Radiator in Rolls-Royce Merlin-Lancaster Power Plant.

rate of flow and of a high rate of heat dissipation are easily seen.

Taking now performance figures from tests on actual oil coolers, we get the curves shown in Fig. 21. These tests have all been made with the relief valve gagged, that is, with full oil flow through the cooling elements.

Fig. 21 (1) shows the rate of unit heat dissipation against air speed at the matrix face, for different air inlet temperatures. The ratio is constant only at low air speeds, when the amount of heat extracted is small and the effect of the stagnant viscous layer of oil relatively unimportant. The falling off of heat dissipation at low air temperatures is most marked. This is due to the increased viscosity of the oil in contact with the metal walls, the consequent thickening of the boundary layer, and an increase in the resistance to heat flow which exceeds the effect of the greater temperature gradient.

Fig. 21 (2) shows the oil outlet temperature plotted against air inlet temperature, for constant oil and air flows and constant oil inlet temperatures. At low air temperatures the oil outlet temperature is seen to rise, so that we have the apparent paradox of increasing oil temperatures with falling air temperatures.

Now the most difficult condition which an oil cooler has to fulfil is usually that arising during climb at low altitudes, in the tropics. Here the heat picked up by the oil has a relatively high value, the temperature of the air is high, and the rate of air flow is low. The cooler will therefore have to be of greater capacity than is required at other engine conditions when, even though the total heat to be dissipated may be greater, the air temperature will be lower and the aircraft speed higher.

However, we have already seen, from Fig.

21, that under conditions of lower air temperature and higher air speed the specific heat dissipation may fall, and it is a fact that unless properly controlled, excessive oil outlet temperatures may result in practice.

4.5. *Control of Oil Coolers.*

It is necessary therefore to provide some form of control of the capacity of the oil cooler, to ensure that the oil outlet temperature may be held between acceptable limits.

The avoidance of excessively low oil temperatures is an obvious matter, usually effected by allowing the oil, when cool, to by-pass the cooling element. The causes of excessive oil outlet temperature, however, are not so readily apparent.

Apart from mechanical causes such as faulty control valves or internal leaks in the cooler which allow some of the oil to short circuit the cooling passages (an extremely common reason), high oil outlet temperatures may be due to one of the following:—

1. *Inadequate mass air flow through the matrix.*

This can arise when the air temperature is abnormally high, or when there

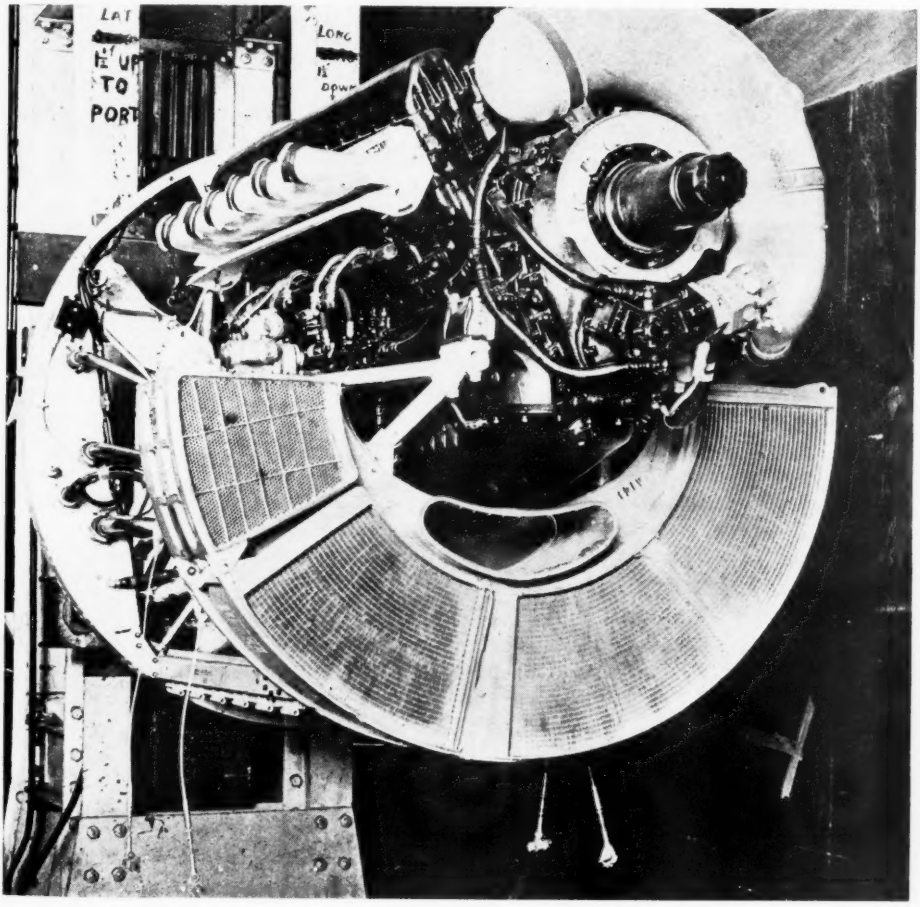


Fig. 17
Oil Cooler, After-coo'er Radiator, and Radiator in Rolls-Royce Merlin-Lincoln Power Plant.

is excessive running of the engine on the ground. The symptoms will be a high oil outlet temperature, not subject to fluctuations, and a low oil pressure drop.

2. *Congealing, or "coring," of part of the cooler matrix.*

This only occurs at low air temperatures, usually well below 0° C. The oil outlet temperature will be steady, and the pressure drop will be high, but not in excess of the by-pass valve pressure. In other words, all the oil will still be flowing through the matrix.

3. *By-passing of the cooler matrix.*

When there is excessive cooling,

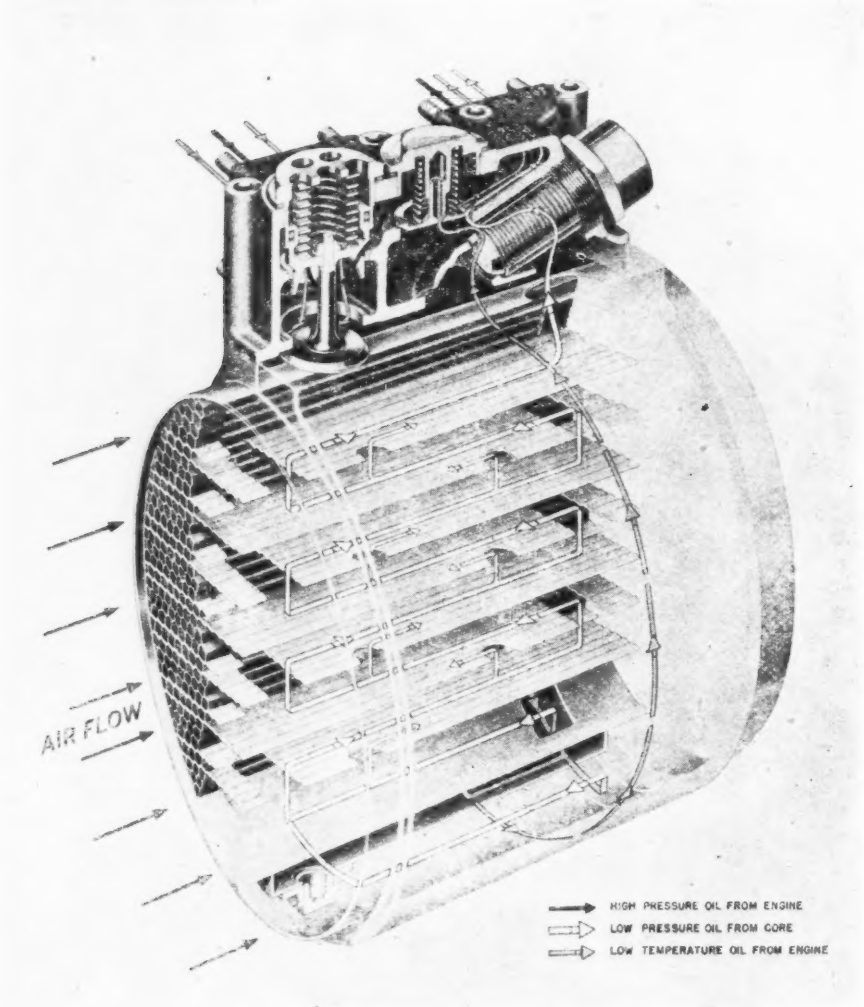


Fig. 18
Skeleton Diagram of Typical Honeycomb Oil Cooler, showing Airesearch "Free-Flow" features and control valve.

which increases the mean oil viscosity to such an extent that the pressure drop through the matrix exceeds the relief valve pressure, oil will by-pass the matrix and will not be cooled. The oil outlet temperature will be fluctuating.

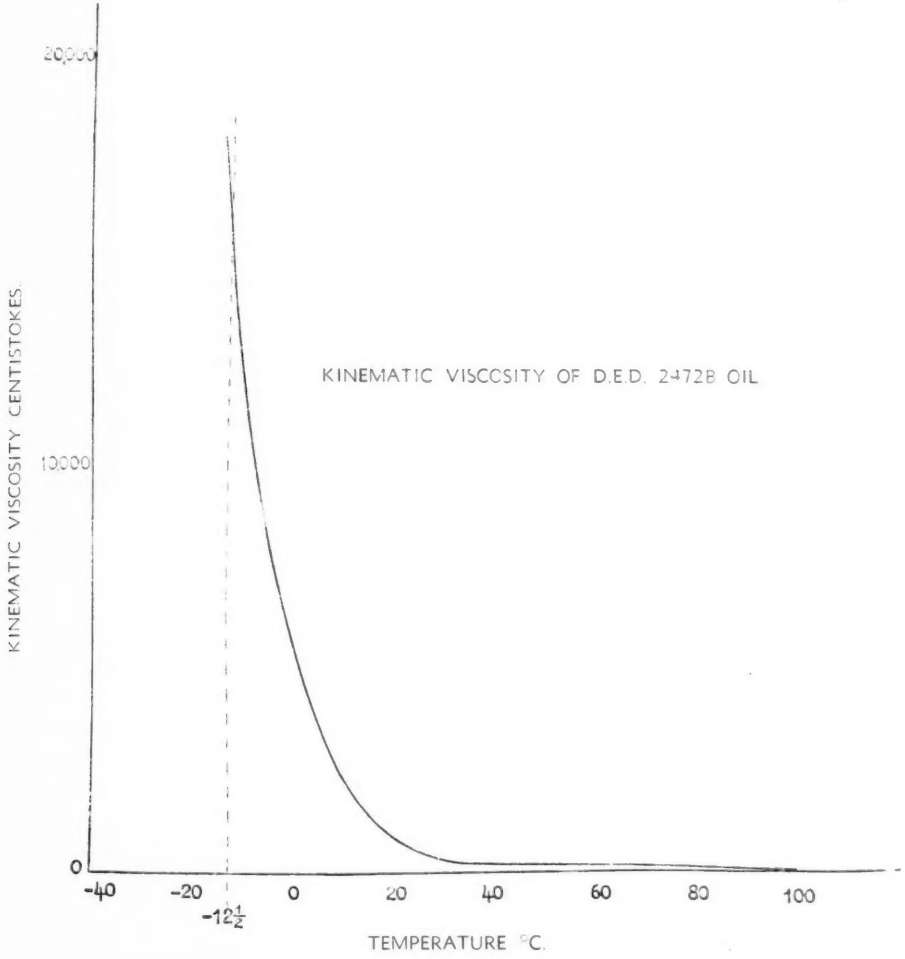


Fig. 19

Curve showing variation of Kinematic Viscosity with temperature. D.E.D. 2472B oil.

4.6. *"Congealing" and "Overcooling."*

These are terms which have been somewhat loosely and interchangeably used. It is now generally accepted that *congealing* means the partial or complete cessation of oil flow in local regions of the matrix. An oil cooler element consists of a large number of passages in parallel, with some measure of cross connection. Even under conditions of uniform temperature and no heat transference the slightest dimensional variation will cause unequal flow distribution, although the pressure drop through all the passages will be the same. When heat transfer is taking place, and the effect is greatly intensified when the rate of heat dissipation is high, there will inevitably

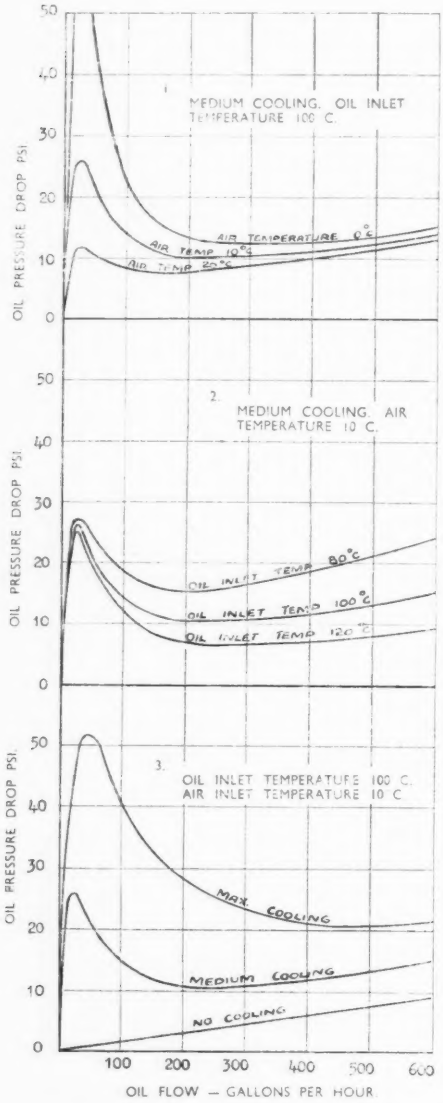
be variations in distribution of air flow, and possibly also of air temperature, across the matrix face. In addition, those passages in which conditions favour heat dissipation, *e.g.*, where the rate of oil flow is lower than elsewhere, will lose more heat, the viscosity will increase and the distribution of flow will be progressively upset until it ceases for all practical purposes in certain parts of the matrix. The oil pressure drop will rise in consequence, but the total heat dissipation will fall and the oil outlet temperature will rise.

The oil pressure drop will not necessarily rise to the point of by-passing, so that full flow of oil will be maintained, albeit through a portion of the matrix only.

No oil cooler is entirely free from a tendency to congeal in some degree, but the tendency to congeal depends largely upon the rate of oil flow, and the variation in oil velocity through different parts of the cooler matrix.

Overcooling, on the other hand, can occur in a well-designed cooler relatively free from congealing. At low air temperatures and high air flows its first effect is to produce undesirably low oil outlet temperatures. As it progresses, the oil pressure drop may exceed the relief valve setting and allow by-passing, which will then cause excessive oil outlet temperatures.

Here then there are two conditions to combat. By-passing can be prevented by increasing the relief valve pressure to the safe maximum. Overcooling, meaning low oil outlet temperatures, can only be prevented by controlling the temperature of the oil leaving the cooler, by means of a thermostatically operated by-pass valve, or by controlling the air flow.


OIL PRESSURE DROP PSI
1. MEDIUM COOLING. OIL INLET TEMPERATURE 100 C.
AIR TEMPERATURE 0°C
AIR TEMP 10°C
AIR TEMP 20°C
2. MEDIUM COOLING. AIR TEMPERATURE 10 C.
OIL INLET TEMP 80°C
OIL INLET TEMP 100°C
OIL INLET TEMP 120°C
3. OIL INLET TEMPERATURE 100 C. AIR INLET TEMPERATURE 10 C.
MAX. COOLING
MEDIUM COOLING
NO COOLING
0 10 20 30 40 50
100 200 300 400 500 600
OIL FLOW — GALLONS PER HOUR

VARIATION OF OIL PRESSURE DROP WITH OIL FLOW UNDER DIFFERENT CONDITIONS.

Fig. 20

Curves showing variation of Oil Pressure drop with oil flow in honeycomb oil coolers (Allfrey: "The Behaviour of Oil Coolers" Ref. 18).

5. OIL COOLER CONTROLS.

There are three practicable ways of controlling oil temperature, by:—

(i) Controlling the oil flow.
(ii) Controlling the air flow.
(iii) Controlling the temperature of the cooling medium.

It is only recently that all aspects of oil temperature regulation have received the

attention due to them, in this country. In America, where extremes of climate are encountered in routine flying during many months of the year, they have had far more consideration, and controlling devices have reached a high stage of development and complexity. Thus in Great Britain there were until recently only three types of control valve—pressure, viscosity, and thermostatic by-pass, while automatic airflow control has been practised only to a limited extent.

American designers have produced such a

FIG. 21.1
HEAT DISSIPATION vs. AIRFLOW.

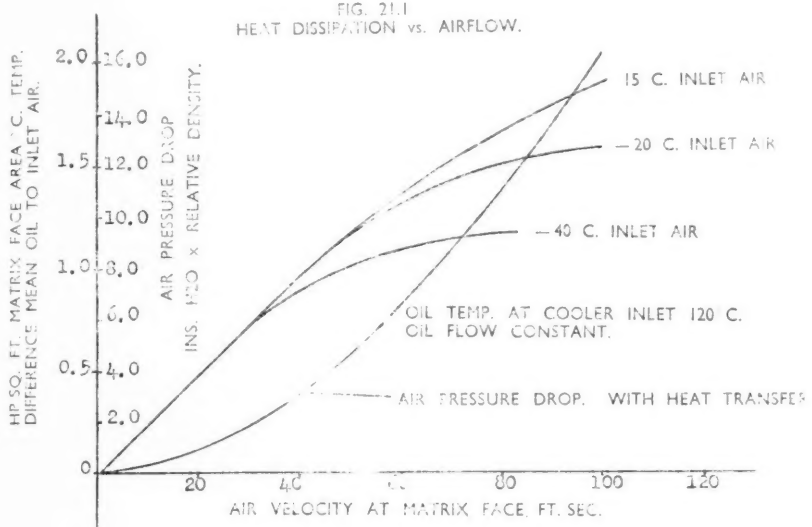


FIG. 21.2
OIL OUTLET TEMPERATURE vs. AIR INLET TEMPERATURE.

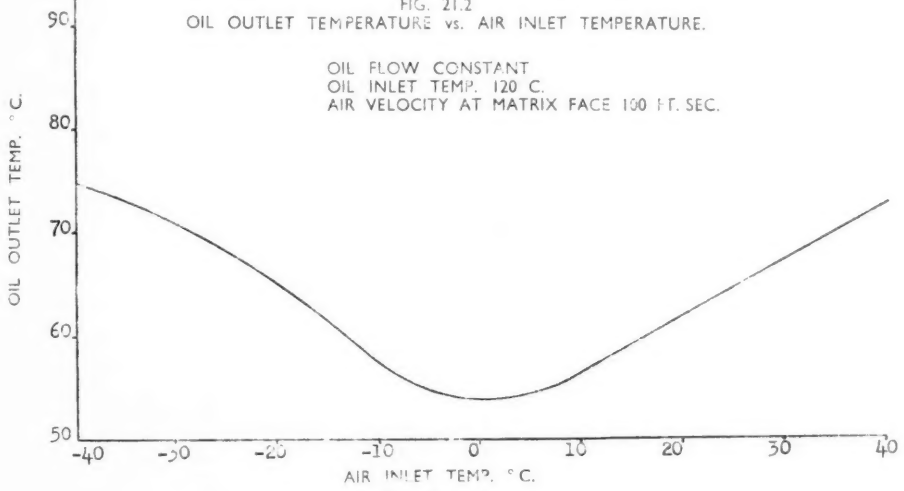


Fig. 21. 1.
Curve of heat dissipation vs. Airflow for an actual oil cooler.

Fig. 21. 2.
Curve showing variation of oil outlet temperature with air inlet temperature, for an actual oil cooler.

multiplicity of types of control valves, pressure-, viscosity-, and thermostatically-operated, and embodying such a variety of functions—thermostatic by-pass, pressure by-pass, and surge protection, with oil- or electrically-operated airflow controls—that it is possible to deal only with one or two representative examples. In addition, prevention of congealing and rapid thawing out have received a great deal of attention, and it is safe to say that few British oil coolers would meet the latest American Army-Navy requirements in these respects.

5.1. *Oil Flow Control.*

(*i*) *Pressure by-pass valve.*

In its simplest form the oil flow control is a spring-loaded ball or poppet valve which lifts when the back pressure of the cooler matrix exceeds a certain value, of the order of 55 lb./sq. in., and allows the oil to pass to the tank, either directly or after having traversed the oil jacket.

This valve has been fitted to large numbers of British oil coolers. It has served only (*a*) to protect the cooler from excessive pressures, though still leaving it exposed to "flick" or "surge" pressures when the engine is started at low temperatures and the cooler and pipe lines are full of congealed oil, and (*b*) to provide quick warming up by by-passing the matrix when it is cold.

It can do nothing, therefore, to control oil outlet temperature save at initial starting, though by setting it at as high a temperature as is safe from the point of view of strength of cooler and pipe lines, high oil temperatures following overcooling and by-passing can be avoided.

(*ii*) *Viscosity By-Pass Valve.*

Far more effective has been the Clarke viscosity valve, Fig. 22, used largely with Rolls-Royce installations. This valve has two elements, a spring-loaded by-pass valve and an automatic control valve. Oil from the engine enters the central chamber of the valve casing, from which a portion of it passes through a filter into the viscosity valve and through a number of long small diameter passages into the by-pass outlet.

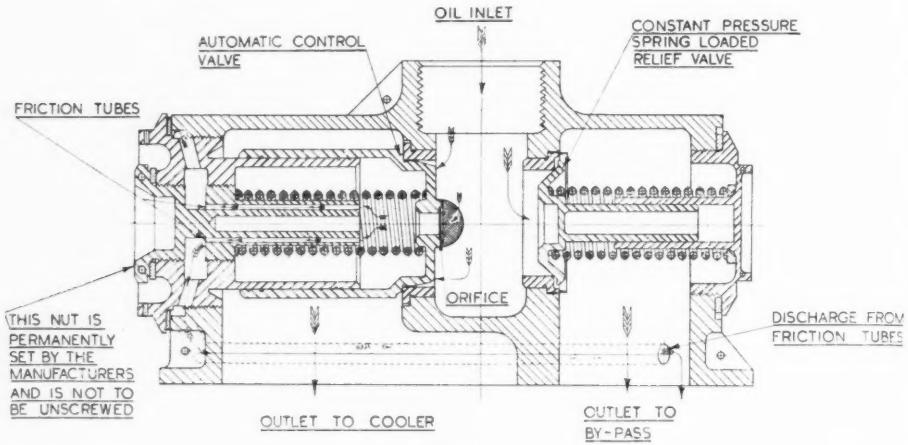

OIL INLET
AUTOMATIC CONTROL VALVE
CONSTANT PRESSURE SPRING LOADED RELIEF VALVE
FRICTION TUBES
ORIFICE
THIS NUT IS PERMANENTLY SET BY THE MANUFACTURERS AND IS NOT TO BE UNSCREWED
DISCHARGE FROM FRICTION TUBES
OUTLET TO COOLER
OUTLET TO BY-PASS

Fig. 22
Clarke Viscosity Control Valve.

When the oil is cold and viscous, the back pressure in the viscosity chamber, plus the spring pressure, keeps the control valve closed, the relief valve lifts, and the main portion of the oil passes directly to the tank.

As the viscosity of the oil falls the by-pass valve begins to close and the control valve to open, so that an increasing volume of oil passes through the cooler. Provided that the

temperature rise through the engine is reasonably constant, this valve will control the oil outlet temperature from the cooler within close limits. It suffers from the drawbacks that under high starting pressures both valves will open and expose the matrix, and that as the relief valve pressure has to match the viscosity valve it cannot altogether prevent high oil temperatures due to overcooling and by-passing.

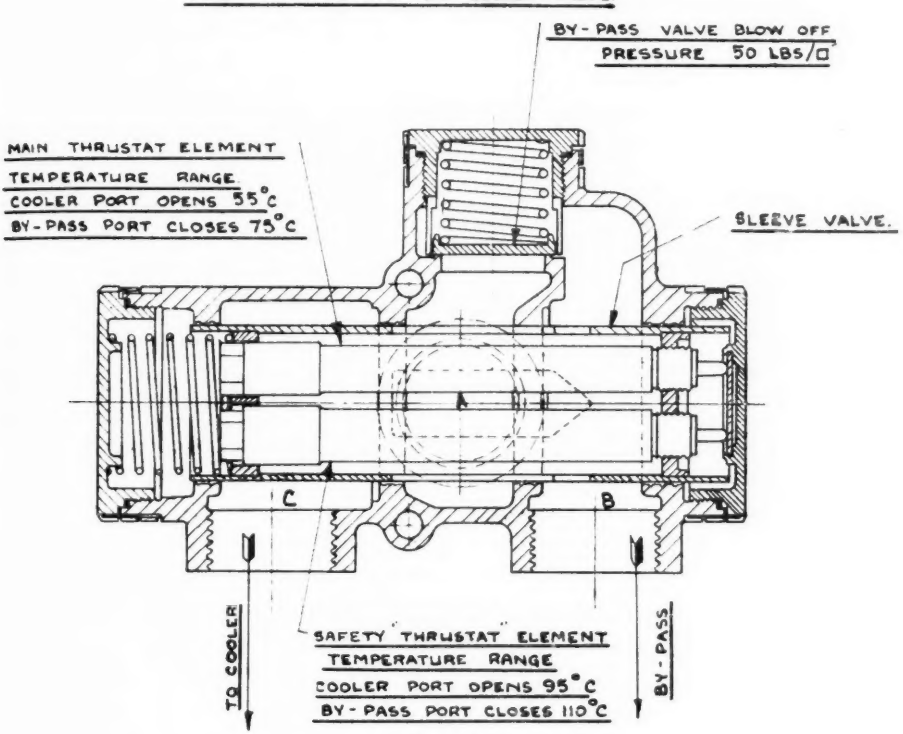


Fig. 23
Thrustat Thermostatic Control Valve

(iii) Thermostatic By-Pass Valve.

Closer control of oil temperatures over a wider range of conditions can be obtained by using a valve incorporating a thermostat, and such valves are now in widespread use. It must be combined with a high pressure spring-loaded relief valve in order to prevent by-passing due to overcooling. There is the additional advantage over the viscosity valve that cool oil diluted with petrol is by-passed to the tank, giving quick warm up and rapid evaporation of the fuel.

In this country it is usually installed to be under the influence of the oil entering the cooler. As it has been used principally with Rolls-Royce engines where the temperature rise through the engine is small, this is quite satisfactory. Where the temperature rise is higher, or subject to variations, a more constant oil outlet temperature is obtained by fitting it in the outlet from the matrix.

Thermostatic valves may employ bi-metallic devices, temperature-sensitive capsules, or special waxes which expand considerably on melting.

In Great Britain the Thrustat valve, Fig.

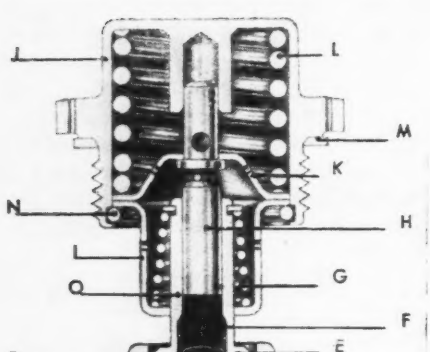


Fig. 24
Vernatherm Thermostatic Element incorporated in UAP D.V.5 Control Valve.

23, which was designed by Mr. T. A. Stern, is now used on all the latest Rolls-Royce installations. It incorporates a ported sleeve controlled by a thermostatic element which consists of a plunger in a cylinder filled with special wax. The valve is illustrated in the by-passing position, with cold oil entering it. When the oil temperature reaches, say, 55° C., the wax begins to melt and expand rapidly, forcing the plunger outwards from the cylinder, and the sleeve valve over to the left, allowing the oil to enter the matrix. The movement is progressive over a temperature range which may be varied by adjusting the composition of the wax.

A second thermostatic element is fitted, set to operate at a higher temperature, so that in the event of the first element losing its wax past the plunger, the valve will still be opened. There is in addition a pressure relief valve, set in this case at 50 p.s.i., which far exceeds the normal working back pressure of Merlin oil coolers and gives an ample margin to overcome congealing or overcooling. While this valve protects the matrix from excessive surge pressures, where the by-pass oil passes through a jacket this will still be subject to these pressures. Although in this country even in winter it is rare to find surge pressures attaining dangerous values, in America it is quite common to encounter temperatures of −40° F. (−40° C.), and it is now obligatory to fit a surge relief valve to all military installations. This valve by-passes the whole cooler and returns the oil directly to the tank. In this country the Napier Company now fits a relief valve between the engine scavenge pump and the oil cooler, which by-passes the oil back to the engine pressure pump.

In America both bi-metallic and fluid-filled sylphon thermostats have been used to a considerable extent, and recently the Vernatherm, which employs the same principle as the Thrustat, has been introduced. This is shown in Fig. 24, from which it will be seen that the wax is completely enclosed in a metal container "B¹," and sealed against leakage by a synthetic rubber diaphragm. In order to increase the speed of response to temperature the wax is heavily impregnated with copper dust.

On expanding, the wax extrudes a synthetic rubber plug, "F," inside cylinder "O," and pushes it against the piston, "H," forcing the valve "A" on to its seat. As the wax contracts on being cooled, spring "G" forces the plug "F" back into position and opens the valve. Spring "L" provides relief in the event of the wax expanding beyond the valve closing position, and it also allows the valve to open when the oil pressure exceeds a predetermined figure.

The illustration shows one particular application. By varying the valve mechanism the valve can be made to open at a given temperature, instead of closing. This element is now embodied in a variety of regulator valves, perhaps the most comprehensive application being that shown in Fig. 25, which is the control valve fitted by U.A.P. to their latest cooler.

This valve incorporates thermostatic by-pass (element "B"), thermostatic surge

relief (element "A"), and pressure by-pass ("E"). When the oil is cold in the cooler, on first starting the engine, element "A" holds the cooler inlet valve shut and disc valve "E" open, and no oil enters the cooler at all, but flows directly from the valve inlet port to the valve outlet port and back to tank. The whole of the oil cooler is thus protected from the starting surge pressure.

As the oil temperature reaches, say, 40° C., the thermostatic element "A" opens the inlet valve and closes the disc valve "E," and oil enters the cooler by-pass chamber, flows through valve "C" and the check valve, back to the tank. Thermostatic element "B," however, does not close valve "C" until the temperature of the oil flowing over it has reached, say, 72° C., by which time the inlet valve will be fully open. The warm oil in the by-pass chamber will now have warmed up the oil in the cooler and flow will be taking place through the case, past the check valve and out.

Should the back pressure of the core exceed say, 55 p.s.i., spring "F" will be compressed, and element "A" and disc valve "E" will be forced to the right, allowing oil to flow directly to the outlet port.

In Fig. 18 was shown, fitted to an oil cooler, an Airesearch control valve. This embodies a spring-loaded surge-relief valve, a capsule thermostatic element, and a check valve. Its method of operation will be readily apparent.

(iv) Oil Flow Control by Internal Leaks.

Much ingenuity has been exercised in the United States to make oil coolers relatively immune from congealing, and at the same time capable of thawing out rapidly with a low maximum back pressure—in some cases not more than about 40 p.s.i. There are two

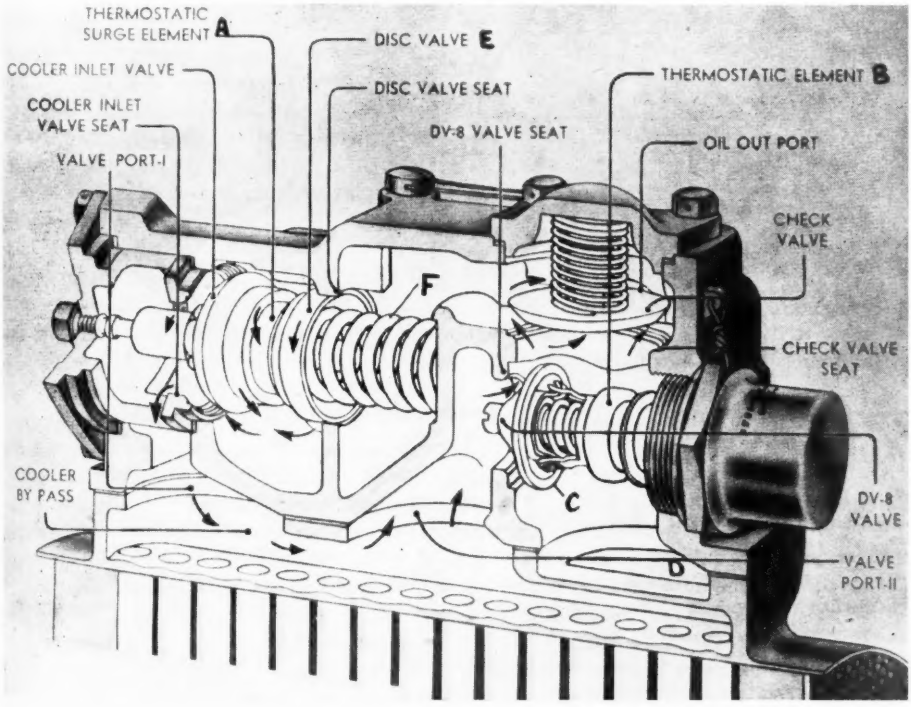


Fig. 25
UAP Control Valve

basic methods, although the actual application of any one of the methods will differ in detail with each size and type of cooler. It is ample evidence of the instability of the conditions obtaining at low temperatures that the final arrangement adopted can only be determined by trial and error methods, sometimes necessitating weeks of work in the cold wind tunnel. Even when a satisfactory layout has been obtained for a 12″ diameter oil cooler, say, and also for a 14″ one, it is not possible to guess the arrangement for a 13″ model of similar general design.

The two basic methods are:—

(*a*) "Diffusion," developed by U.A.P., which involves the removal of numbers of tubes from the matrix and plugging the ends, together with the provision of small bleed holes in the inner casing of the jacket. The tubes are removed in a more or less regular pattern, permitting an easy flow of oil through the core, and arranging the easy flow paths to give maximum heating effect to the remainder of the matrix. The diffusion holes allow a controlled leakage of hot oil from the jacket into the matrix. See Fig. 26.

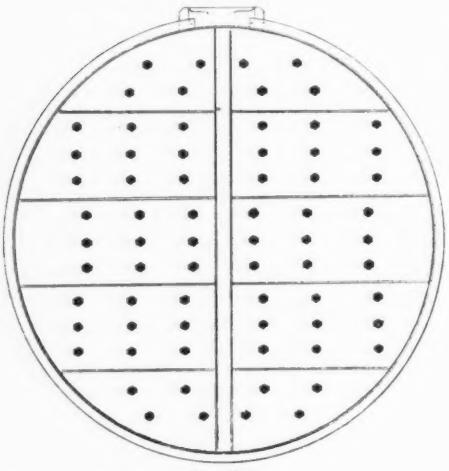


Fig. 26
Approximate Disposition of Dummy Tubes in 12″ Diameter Jacketless UAP Oil Cooler.

(*b*) Free-flow, developed by Airesearch and others. The oil cooler in Fig. 18 shows this feature. Auxiliary ports are provided in the baffles, and the blocks are spaced away from the baffles by packing strips, as shown in Fig. 27. Additional dummy baffles may also be fitted.

It is necessary to strike a nice balance to obtain freedom from decongealing, and rapid thaw out, without sacrificing too much performance under warm conditions.

5.2. *Air Flow Control.*

Airflow control is used to prevent overcooling, to delay the onset of congealing, to accelerate warm-up on starting, and to reduce cooling drag.

In its crudest form it has been practised in cold weather conditions by fitting blanking plates on to the cooler matrix face. This method is only applicable when operations are confined to a given locality. Too often blanking plates are applied without thought to the cooler characteristics. Since the object is both to prevent congealing and to delay the opening of the relief valve, the blanking plate should cover the hottest part of the oil cooler, that is, the first passes. The pressure drop is lowest here, so that a high pressure is maintained at the entrance to the passes more prone to give trouble, together with a high oil temperature. It is essential, too, to ensure that the blanking plate does not cover a portion only of any particular pass, as then the balance of flow through that pass would be upset and congealing would be encouraged.

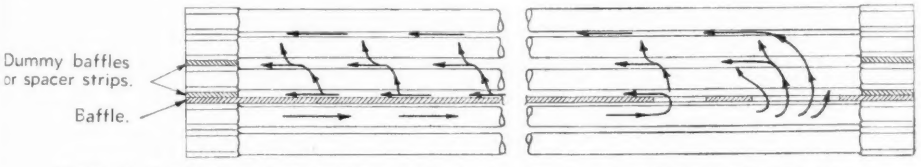
Far more control is obtained by shutters of venetian blind form, fitted at the rear of the cooler. These are not good aerodynamically, but they possess the merit of being with the cooler always, and rendering it relatively immune from the effects of bad installation.

Aerodynamically more correct, and capable of giving a considerable saving in drag at medium and low airflows, is a flap at the duct exit, which may be controlled by any of the methods applicable to shutters.

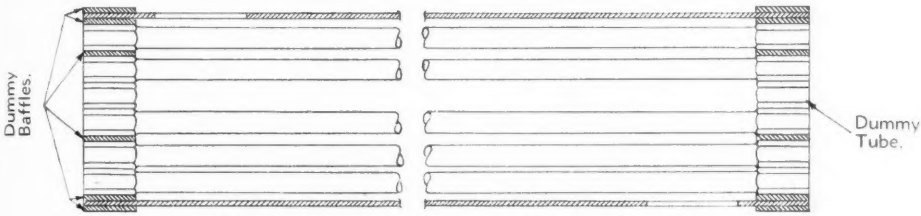
Control of shutters is effected by means of the pressure drop through the cooler, or the

temperature of the inlet, or the outlet, of the cooler, or by the viscosity of the oil. The motive force may be the oil pressure, or a hydraulic or electro servo. Pressure drop is usually employed in Britain, oil temperature in America.

American manufacturers have produced valves incorporating surge relief, pressure by-pass, and, in addition, shutter-actuating pistons controlled thermostatically by the oil outlet temperature. In the example shown in Fig. 28, by Airesearch, a bi-metallic thermo-



SECTION THROUGH BAFFLE AND DUMMY BAFFLES OF AIRESEARCH FREE-FLOW OIL COOLER



SECTION THROUGH PASS OF YOUNG RADIATOR.

Fig. 27
Methods of obtaining Controlled Internal Leaks.

static element operates a rotary valve controlling oil flow to the servo piston. There is a pressure override valve to close the shutters when the outlet oil is hot through the matrix being congealed.

Later controls produced by this company act through an electrical actuator under the influence of a thermostat. Fig. 29 shows diagrammatically the layout of such a scheme, in which the flap is opened progressively in phase with temperature increase.

5.3. *Control of temperature of the cooling medium.*

This is practised to some extent in the latest Rolls-Royce "series" installations, where excessively low air temperatures are avoided by installing the oil cooler behind the coolant radiator of a liquid-cooled engine. In another form, engine coolant is used instead of air, and with this scheme a closer measure of cooling temperature control is possible.

Neither of these methods is feasible with air-cooled engine installations. Any other means of controlling the temperature of the air would probably lead to an impossible complexity of ducting and controls, although with the increasing adoption of supercharger intercoolers and cabin heaters there is here an available supply of warmed air which might be utilized.

6. CONSTRUCTION AND MANUFACTURE.

The simplest oil cooler of the present day is without doubt that of cylindrical form produced by American manufacturers. It has been well-designed from the production point of view, containing the absolute minimum of riveted joints and added stiffeners. Essentially it consists of a honeycomb matrix formed by dipping the hexagonal ends of the tubes into soft solder, and, usually, a double-walled

casing. The casing comprises one purely cylindrical part, of single thickness, with an outer spinning, the two being joined together by silver soldering or brazing.

The matrix elements are made by assembling the requisite number of tubes together in a jig, with tubes with specially shaped ends, and parallel tubes of D cross section, to make up the required outline, fluxing, and dipping the ends in soft solder. The blocks are next inserted in the casing, and the peripheral joints are made by hand, using a small pencil flame of a coal-gas/air torch, and soft solder. Fig. 30 shows the elements of such a cooler.

In this country it is the usual practice to have the baffles of flat sheet, forming part of the structure of the casing. This necessitates pentagon-ended tubes to make a good baffle joint. In America, in the cylindrical coolers, it is more usual to assemble the baffles in the block with the tubes and dip them all together. The baffles are corrugated to fit the hexagonal tube ends. In these coolers the fit between the flanged ends of the baffles and the inner casing is depended upon to maintain freedom from leaks between adjacent passes, but this method is not altogether desirable, as this object is not always achieved.

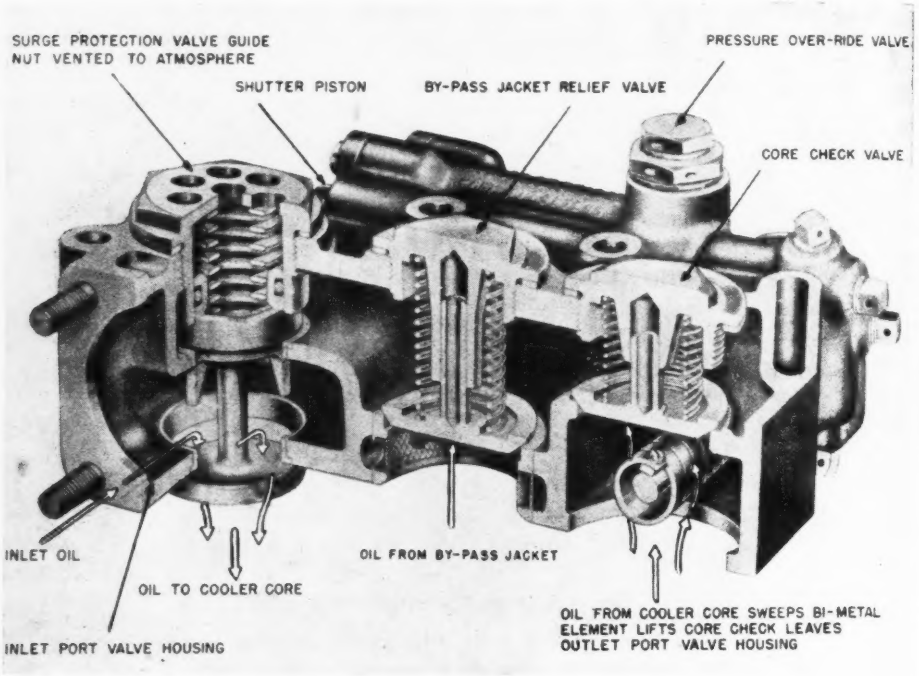

SURGE PROTECTION VALVE GUIDE NUT VENTED TO ATMOSPHERE
SHUTTER PISTON
BY-PASS JACKET RELIEF VALVE
PRESSURE OVER-RIDE VALVE
CORE CHECK VALVE
INLET OIL
OIL TO COOLER CORE
OIL FROM BY-PASS JACKET
INLET PORT VALVE HOUSING
OIL FROM COOLER CORE SWEEPS BI-METAL ELEMENT LIFTS CORE CHECK LEAVES OUTLET PORT VALVE HOUSING

Fig. 28
Airesearch Control Valve, incorporating airflow control.

6.1. *Materials.*

Honeycomb tubes are now invariably made from oxygen-free high conductivity copper of high purity, or "cap copper," which contains 3% zinc. They are produced either by drawing or by impact extrusion. The accuracy of size and shape of the hexagon ends is important to ensure a well-fitting nest of tubes.

This copper is extremely tough and ductile. Its readiness to take solder is a valuable asset, but the solubility of copper in tin is a point to

be guarded against by controlling the maximum time and temperature of dip in the solder bath.

With few exceptions the casing and baffles are made from cold rolled brass sheet, usually "basis" brass containing 63% copper and 37% zinc, cartridge brass (70/30) being used for the parts which require more forming or where strength is less important. These materials are ductile and easily formed, but

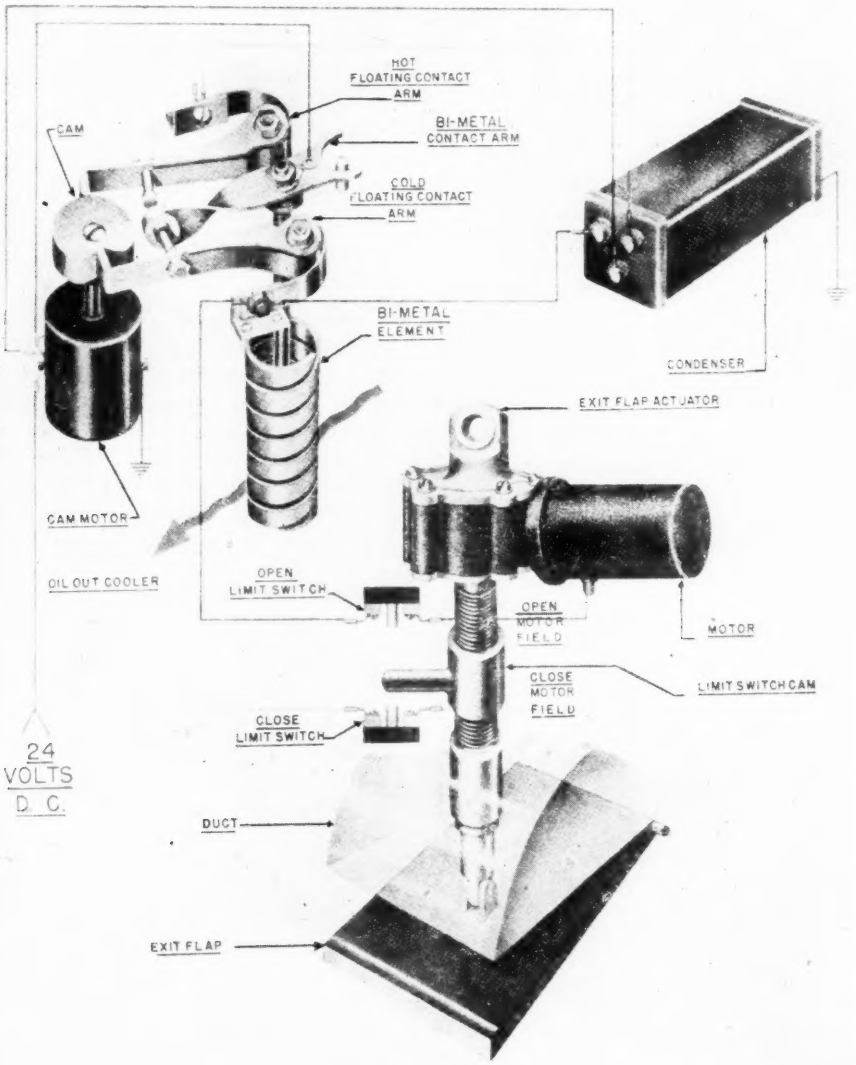


Fig. 29
Airesearch Floating Control for Exit Duct Flap.

are prone to "season cracking," in areas which have been subjected to stress, as in forming without subsequent annealing, when exposed to atmospheric corrosion. Cracking may also occur due to tin penetration when solder is applied to parts which have not been annealed.

The principal jointing material, soft solder, has been greatly maligned, most people looking upon it as anything but an engineering material. This is probably because of the relative ease with which the inexperienced amateur can use it for kitchen repairs. It is an attitude which unfortunately has spread to industry, and the author has come across many glaring examples of complete failure to understand its properties.

Soft solder is an alloy of lead and tin, possibly with the addition of a little antimony or silver. In view of the prevailing ignorance there need be no excuse for reproducing the phase diagram in Fig. 31. This shows that there are two important temperatures for each alloy—the liquidus, when the solder is fully molten, and the solidus, when it becomes completely solid. Between these two temperatures the solder is pasty. When making a soldered joint it is essential to ensure that the mating parts and the solder reach a temperature in excess of the liquidus, as otherwise the solder will not wet the metal and form an alloy with it, and the solder in the joint will be pasty, and weak on cooling. A superheat of about 100° C. above the liquidus gives the strongest joint (Ref. 19). Also, on no account must a joint be disturbed as the solder is cooling through the pasty range, as otherwise a defective joint will result.

Before the war the most commonly used solders were the 50/50 alloy, or one having 45% tin, 2½% antimony, 52½% lead. These solders were easy to manipulate, as they had a narrow pasty range and were therefore less liable to be disturbed on cooling.

Owing to the need for economy in tin, however, it has been necessary during the past three years to use alloys containing about 30% tin, with a greater pasty range. The solidus temperature for all solders is practically constant at 183° C., but the economy solders require a higher temperature to ensure that the liquidus has been passed, and greater care during the longer pasty range.

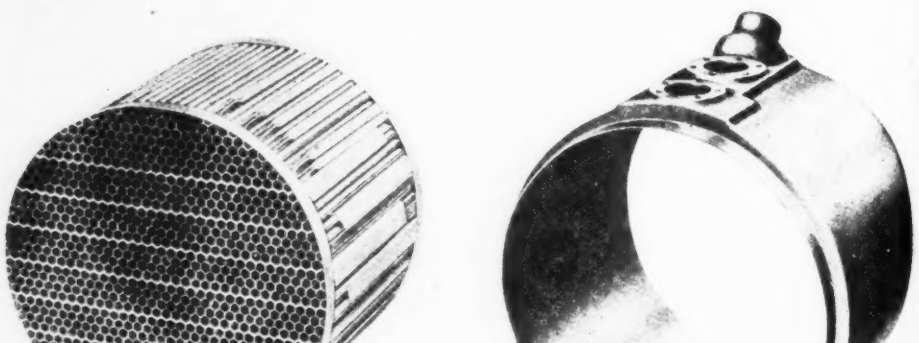


Fig. 30
Typical American Oil Cooler Core and Casing.

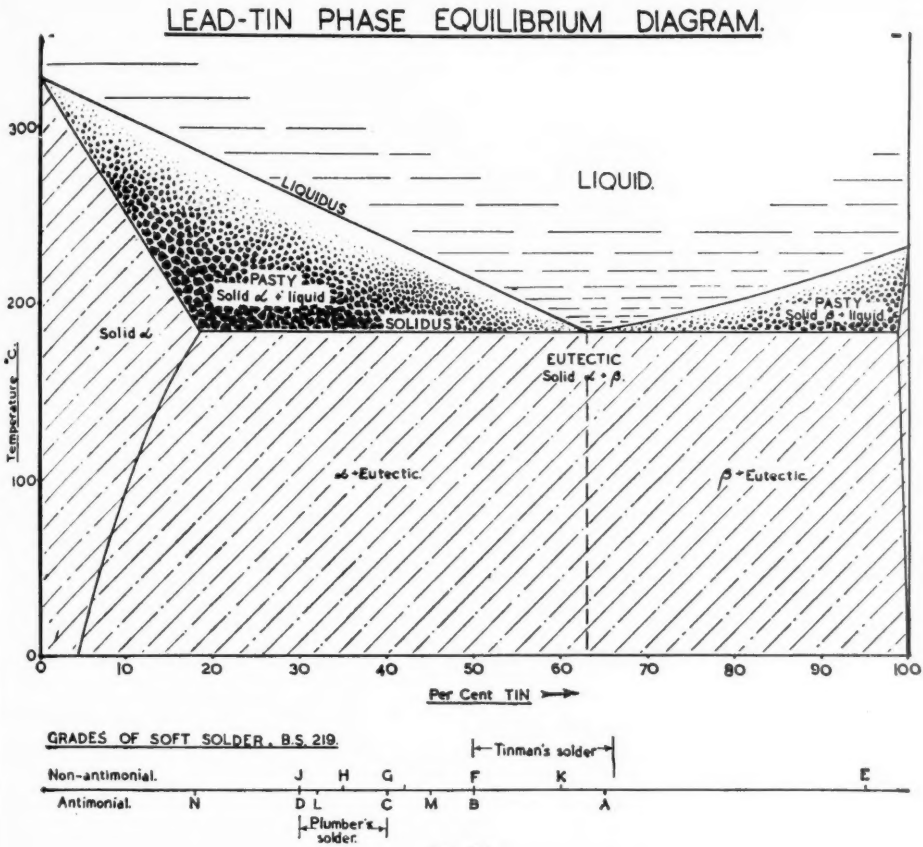


Fig. 31
Phase-Diagram for Lead-Tin Solders.

Soft solder will only "wet" parts which are absolutely clean, and in order to ensure this the parts are treated with a flux immediately before the solder is applied. The action of fluxes is not fully understood, but their main purpose is to remove grease, dirt, and oxide film, and to promote wetting.

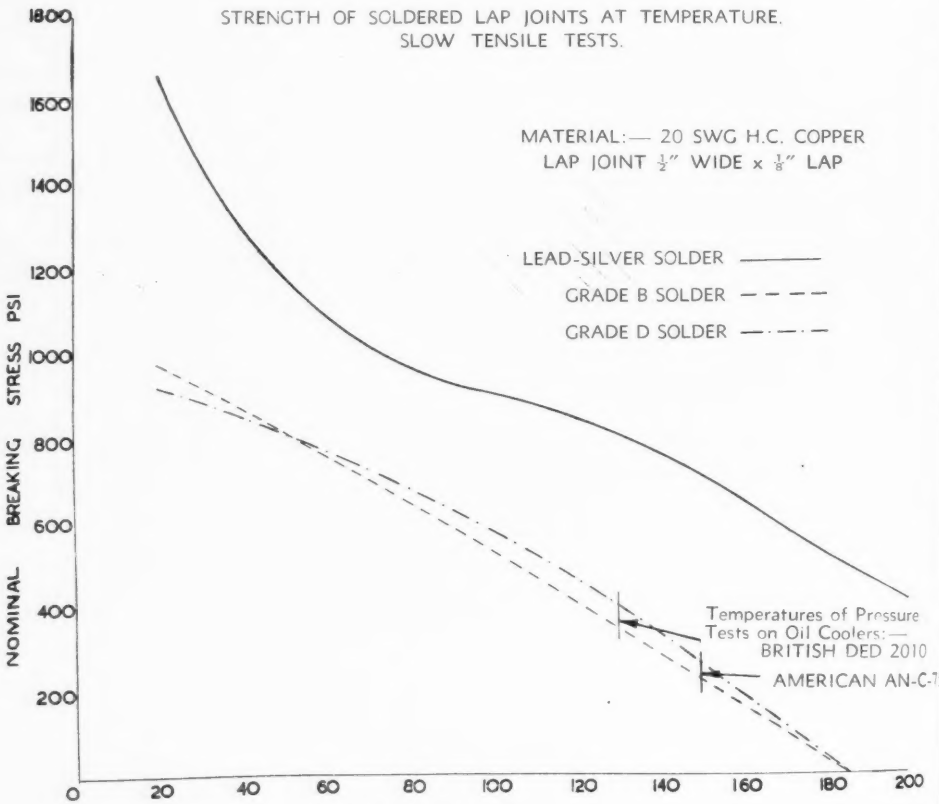
Unfortunately the most active fluxes are corrosive and, after heating, their residues are relatively insoluble in water, and still corrosive. The flux normally used in oil cooler construction is D.T.D.281, a mixture of the chlorides of zinc and ammonium, in water. This, and its dried residue, are soluble in dilute (2%) hydrochloric acid, which is accordingly used to wash the work after soldering. This is usually followed by a fresh water wash, then one using soda ash solution, with a final fresh water wash.

Solder has considerable surface tension, and in a joint having a gap of, say, .003" to .007", a considerable rise of solder takes place due to the capillary effect.

The strength of a soldered joint, properly made, depends upon the composition of the solder, the metals being joined, and the thickness of the solder film. Maximum strength for any given combination is obtained when the thickness of solder is of the order of .005" (Ref. 19). Unfortunately, this requires greater accuracy of pressings than is usually encountered in the industry.

With an initial melting point of 183° C., it is to be expected that the strength of soft solder will be falling off rapidly at the temperatures reached by lubricating oil. Fig. 32 shows the strength of two solders in lap joints in copper $\frac{1}{2}$" wide by $\frac{1}{8}$" lap, broken in tension (*i.e.*, combined shear and tearing in the solder film) under conditions of slow loading, which approximate to creep tests carried out over a period of from ten to twenty days. These give a much more nearly true picture of the actual strength of solder under service conditions than do rapid tensile tests. The curves are based on work which has been carried out by the British Non-Ferrous Metals Research Association for M.A.P., in connection with the latter's development programme on oil coolers and radiators.

Although the stress figures may appear low when compared with high tensile aircraft materials, it must be remembered that the area of a soldered joint is usually relatively great (about $\frac{1}{2}$" wide) compared with, say, the useful area of a riveted or spot-welded joint.



SOLDERS

TYPE	COMPOSITION Sn	Pb	Sb	Ag	SOLIDUS	LIQUIDUS
LEAD-SILVER ...	1.0	97.5	—	1.5	310°C	313°C
BS. 219 GRADE B.	50	47.5	2.5	—	185°C	227°C
BS. 219 GRADE D.	30	Rdr.	1-1.7	—	185°C	248°C

Fig. 32

Strength of Soldered Joints at Temperature. Slow Tensile Tests.

For oil coolers dealing with oil at inlet temperatures up to 150° C., there has been in my own experience no indication that soft solder is not completely adequate for the duty imposed. In confirmation of this it is American practice to carry out long endurance tests at this temperature.

With air-cooled engines, however, there is a trend towards still higher oil temperatures at the outlet from the engine, and to meet these it will be necessary to adopt jointing alloys of a higher melting point. A great deal of work has been carried out by the B.N.F.M.R.A., and in Fig. 32 is shown a curve for one of these, a high lead-silver solder which melts at 310° C. Used with a suitable flux, this is giving satisfactory results in service, and it has the big production advan-

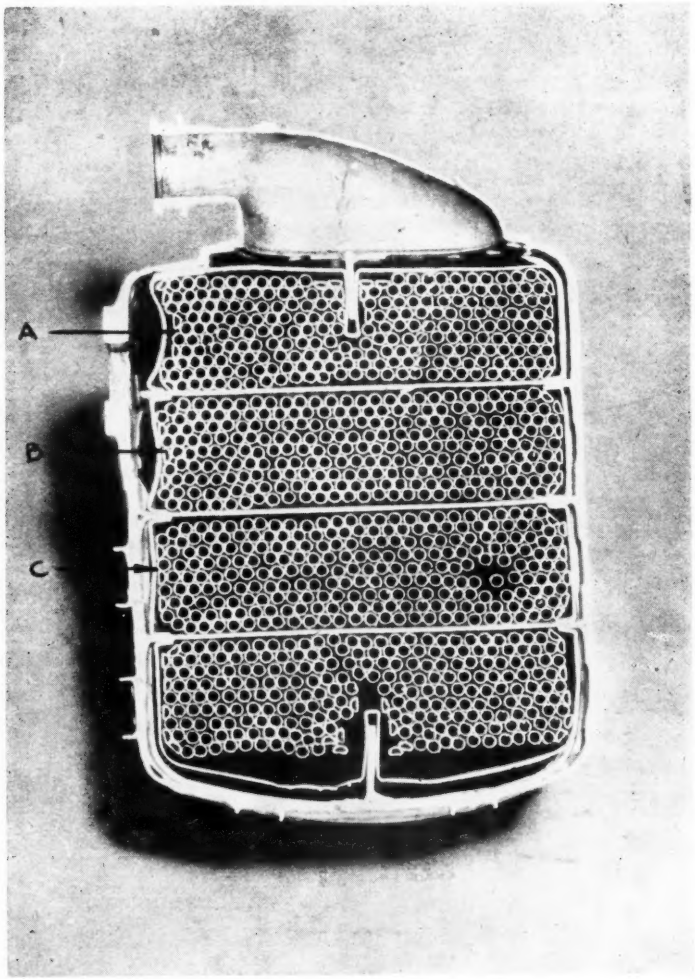


Fig. 33
Section through oil cooler after starting at extremely low temperature, in Canada.

tage of requiring only a small change in handling technique.

7. SERVICE TROUBLES AND THEIR REMEDIES.

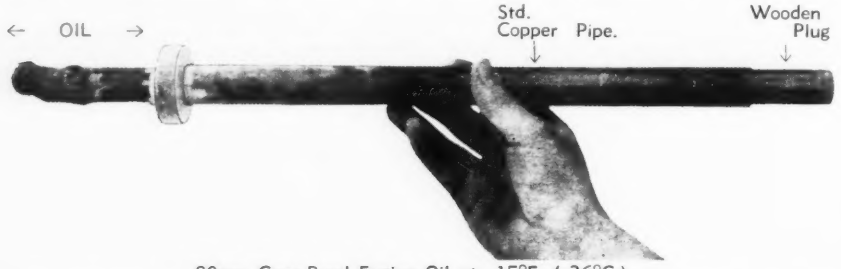
For the past year I have been carrying out on behalf of M.A.P., numerous investigations into the causes of failures of oil coolers of all types used by the R.A.F. This experience has been most valuable, and in what follows are given the general conclusions which so far it has been possible to draw. These are based on all types of British oil coolers, and the subject matter of this section applies generally, without distinction or discrimination, to all makes and designs which have been in use.

The low standard of reliability of oil coolers in service during the war is too widely known for it to be a military secret. Initially this state of affairs was enjoyed more or less equally by the Americans and ourselves, but they have succeeded in the past two years in raising the standard of reliability of oil coolers to a most acceptable level.

A purely personal opinion is that our tardiness in tackling this problem has been due to a large extent to the emphasis given to the more highly technical aspects of oil cooling, so that it was natural, for quite a long time, to ascribe all the troubles to congealing or coring or high back pressures.

As in many other cases when trouble arises, there has perhaps been too great a tendency to look only to theoretical or functional factors to account for failures, when often the main or basic cause has proved to be much simpler, and has been found in the material or the manufacture.

This is not to say that the other factors are not important. True diagnosis must take into account all possibilities, and sometimes it is found that conditions of operation or duty have an effect second only to poor design or unsatisfactory manufacture. When there are overriding causes of failure, such as defective workmanship, however, it is almost impossible by elementary statistics to draw correct conclusions regarding the part played by the duty imposed upon the component, or, in common parlance, by the "working stresses."



80sec. Grey Band Engine Oil at -15°F. (-26°C.).
Note.—Oil was frozen in a 1" i/d copper pipe. A load of 50 lbs. was required to force the column of oil out of the 16" length of pipe, at a speed of 4 inches per minute.

Fig. 34
What happens to oil at low temperatures.

Having examined some hundreds of failed British oil coolers, I can say at once that defective manufacture has been the greatest single cause, with poor detail design second. It is a fact that in all the cases examined there has not been one where an initially sound joint has failed. In most cases the surprising thing is that failure had not occurred sooner.

The chief exception to this is oil cooler failure through excessive pressures imposed when starting up with all the oil in the cooler and pipe lines frozen. This has occurred rarely in this country, but more commonly in Canada and the United States (see Fig. 33). It can be overcome by fitting a surge relief

valve, which is now obligatory in the United States, where oil in the condition shown in Fig. 34 is the rule in winter. (Ref. 20.)

When all these defects have been eliminated and the components achieve a considerably longer service life, then it may be that frequent indulgence in congealing will put a period to the life of a given design of oil cooler, since this gives rise not only to higher than usual pressures, but to a steep temperature gradient between the operating and the congealed areas, a difference of 100° C. having been observed at points half an inch apart (Ref. 21). Design from the heat transfer aspect has reached a fairly high plane, however, and it is felt that when we have eliminated the present defects there will be little cause for complaint.

7.1. *Examples of Defects.*

A honeycomb oil cooler, 16" diameter, will have incorporated in it over 3,000 thin-walled tubes, having a total length of over 1,000 yards. The soldered joints will have an aggregate length of about 80 yards. In some designs, especially those with flat-sides, the number of rivets used may exceed 1,000. The important part assumed by the detail manufacture of the oil cooler should therefore be obvious, yet we have it on the authority of Air Commodore F. R. Banks (Ref. 22) that existing technique "has been very crude."

The most common source of oil cooler leakage, in British designs, occurs at the rivets used in attaching stiffeners to casings, or in joining the jacket seams. While these are not invariably dangerous, they are a nuisance for which there can be no justification from the point of view either of designer or producer. To aim at producing a pressure vessel which is as full of holes as may be, stopped up with rivets which are all too often badly formed or inadequately soldered is, to say the least, doing it the hard way.

Another trouble of widespread but not always serious nature is due to leaks along the joints between the hexagon-shaped ends of the tubes. These usually occur locally and

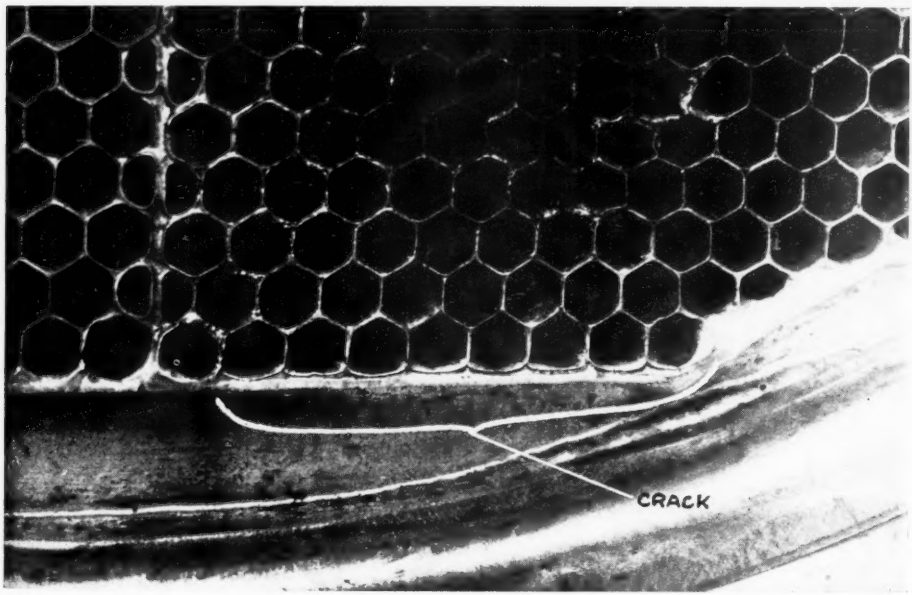


Fig. 35
Example of Block Casing Joint Failure.

are frequently but not always, of small extent. They are caused by poor fitting together of the tubes, or inadequate cleaning and fluxing, or draining away of the solder from the space between adjacent hexagon flats, when this space is too wide to hold the solder under the conditions of bath temperature and depth of dip which obtain.

Far more serious failures, permitting rapid loss of oil and causing early engine failure, are those which occur in the joint between block and casing, or between block and baffles.

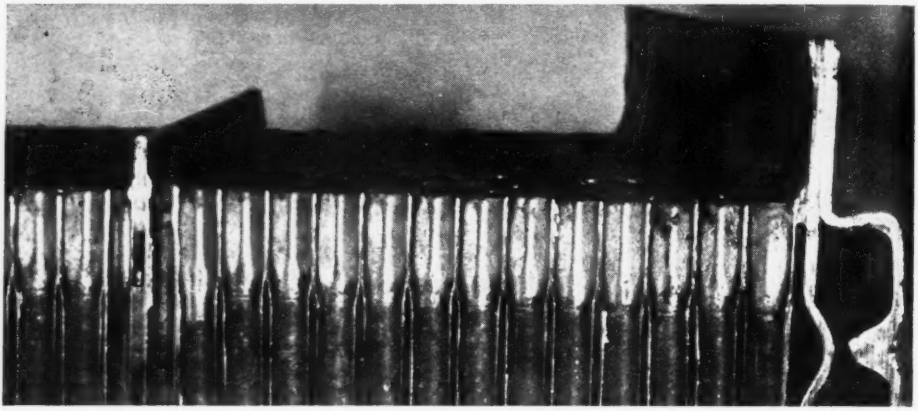


Fig. 36
Section of Honeycomb Oil Cooler, showing scanty joint between block and casing.

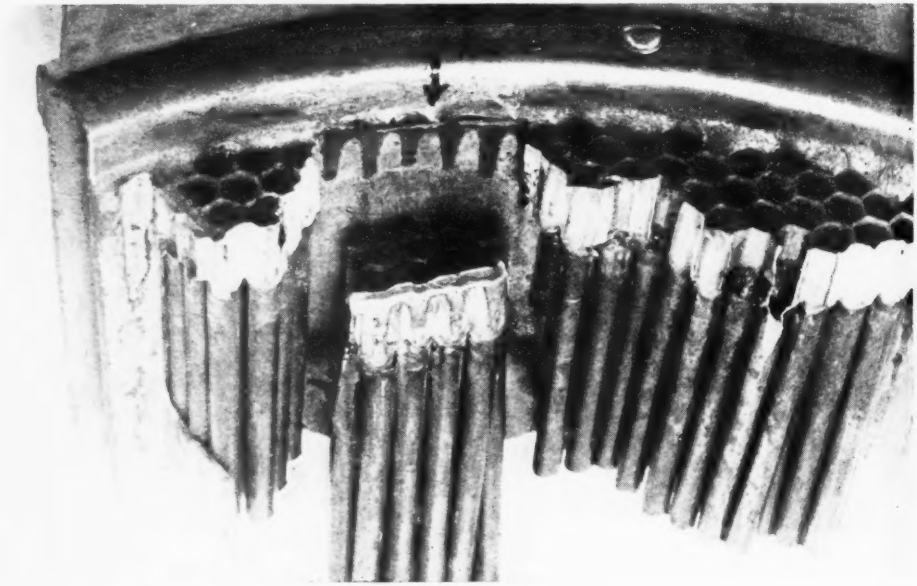


Fig. 37
Typical peripheral joint failure opened up for examination.

Fig. 35 shows a failure in a block-to-casing joint which was responsible for an engine failure through loss of oil. The crack occurred and the engine failed after a total life of three hours.

Fig. 36 shows a cross section through a typically unsatisfactory peripheral joint, and Fig. 37 shows such a joint, which had actually failed, opened up for examination.

In Fig. 38 there are sections of three adjacent passes, which had never been joined to their baffles. The scantiness of the solder seal

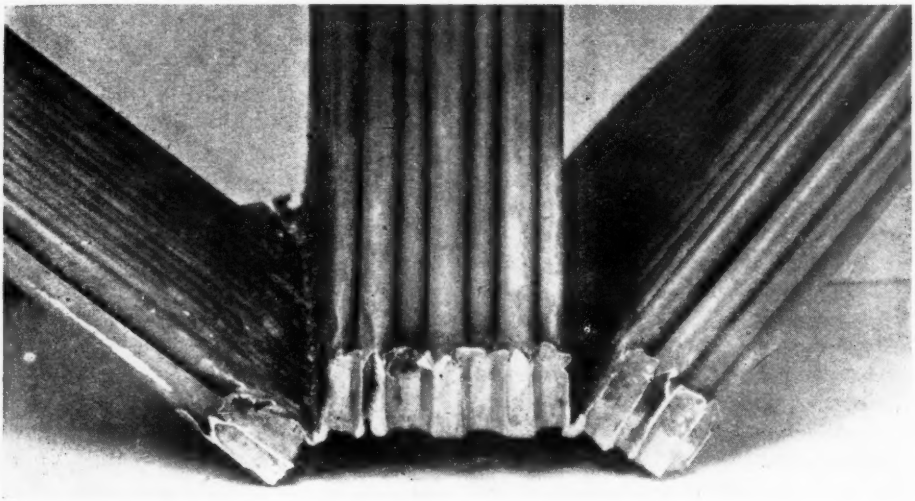


Fig. 38
Tubes from three adjacent passes, showing lack of bond to baffles.

Fig. 39
Joint between inner and outer casings opened to show poor fit and lack of solder.

is alarming.

Fig. 39 shows a common cause of leak caused by inadequate soldering of the joint between inner and outer casings, and Fig. 40 is an example of a poorly fitting inner casing which has not allowed a proper joint to be made. A leak has occurred between matrix and jacket, and an excessive oil outlet temperature has resulted.

By way of comparison, Fig. 41 shows the peripheral joints of an American oil cooler, taken at random from stocks in this country. These can be classed as perfectly satisfactory, since the sections where the joint is short are still twice as deep as some of the bad examples, while they are so narrow that they are adequately supported by the adjacent areas and failure cannot occur.

Another cause of failure, fortunately not so common as the others, comes about through the perforation of some of the tubes by corrosion due to residual flux which has not been removed by washing after the last soldering operation.

In the same category is bursting of tubes by frozen rain water, which may lodge in a tube whose rear end may have been blocked by solder during manufacture or repair, or by dirt or foreign matter during use. Defects in the tubes themselves are responsible for a small number of failures, while they have a secondary effect due to only partially successful efforts at repair when a defective tube is replaced, in a new cooler, before delivery. These have been reduced to a minimum in America, where the oil cooler manufacturer subjects every tube to a pressure test before embodying it in a matrix assembly.

7.2. *Basic Causes of Failure.*

Speaking generally, the great majority of present troubles can be ascribed to non-observance of any of the following essentials:

1. Good fit between mating parts.
2. Absolute cleanliness of the joint faces.
3. Proper fluxing of the joint faces.
4. Adequate heating of the metal and the solder.

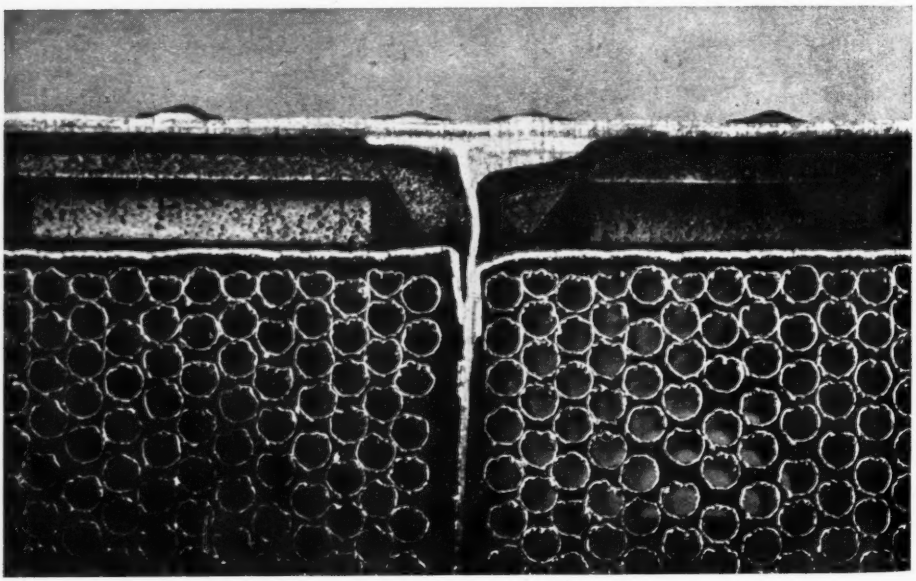


Fig. 40

Section showing poor joint between inner casing and baffle, permitting internal by-passing.

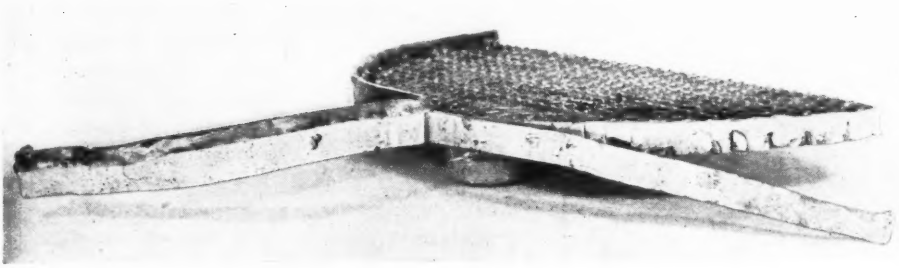


Fig. 41
Typical American oil cooler section, showing sound peripheral joints.

There is an additional cause of failure which applies particularly to oil coolers having casings of fairly heavy steel, sometimes unsymmetrical in design, in which the tubes are assembled and dipped, instead of being dipped in a jig and soldered into the casing by hand. Here the higher specific heat of steel as compared with brass means that for the same time and temperature of immersion in the solder bath there is less assurance of attaining correct soldering temperature. More important, the difference in coefficients of expansion of the steel casing and the copper block means that the matrix will be constrained and the tubes packed more closely together when the assembly is heated by immersion in the solder. On cooling, the matrix will contract more than the casing and will tend to draw away from it, with grave risk of cracking in the peripheral joints. If in addition the casing expands unsymmetrically, this tendency will be increased. I have encountered one case in which the carcass of a steel oil cooler actually contracted in one direction when it was heated, due to its asymmetry. Finally, steel does not "wet" nearly so easily as brass. In the cold-rolled close-annealed form it is difficult to degrease it thoroughly as the rolling lubricant, which may be partially burnt on, is extremely adherent, and the tinning quality may vary from batch to batch.

7.3. *Obviation of Defects.*

It will be interesting now to consider the immediate underlying causes which give rise to the troubles just described, and to outline the steps which are being taken, or which still remain to be taken, in order to eradicate them.

(*i*) *Rivet Leaks.*

The soundest way to avoid rivet leaks is not to have any rivets. Unfortunately, even in cylindrical designs where their employment could well have been avoided, they are used to a considerable extent. Even if we do condone them in pressure containers, they are not the most economical means of attaching parts together, and when the operative knows that subsequently they are going to be hidden under a coating of solder, he may be tempted to cobble rather than rivet them, especially when the fit of the parts is poor, and riveting consequently difficult. Added to this, there may be considerable difficulty in making a seal with solder. Electro-tinned rivets are used, but unless the coating is sufficiently thick it will not be possible to store the rivets for any length of time, or the whole of the tin coating will become oxidized and soldering will then be impossible. Moreover, the drilled holes in the casing will usually be untinned unless a most laborious operation is introduced, and the very act of riveting precludes the subsequent entry of flux, to enable the tin on the rivet to bond with the metal of the shell, in the hole.

It is far better to design a casing to be adequately stiff without extraneous support. This can be effected as in American cylindrical designs by having the casings sufficiently thick

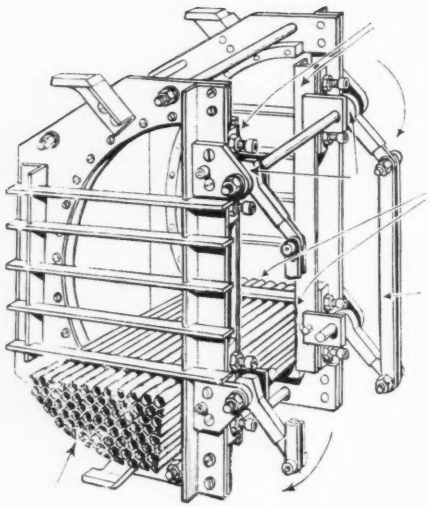


Fig. 42
Gated Dipping Jig Evolved by Delaney-Galley, Ltd.

to withstand the maximum test pressure (100 p.s.i.) without distortion, and by brazing or silver soldering the two casings together. In the oval designs the outer casing is corrugated, and seam-welded to the inner casing, making a rigid construction.

Flat-sided oil coolers can be made in the same manner, or by dimpling both inner and outer casings so that the crests of the dimples will touch, when they can be spot-welded.

It is to be regretted that although the progress made in the technique of resistance welding has been so great of recent years, there is still the most widespread ignorance of even the simple fundamentals of spot welding such an easy material as mild steel. The fact that this is the technique of the blacksmith brought under automatic control does not appear to be realized, and many cases have been seen of faulty spot-welds due to insufficient temperature, or pressure, or both, or either applied at the wrong time.

(*ii*) *Matrix Face Leaks.*

When the tubes are nested together in the dipping jig, many of the adjacent hexagon flats will be in contact, but there will usually be small interstices due to the radii at the corners. It is commonly found on opening up these joints that the solder has penetrated most of them, but that small islands of bare copper exist in the centres of some of the flats. Provided that these are not extensive, they are unlikely to cause trouble. Where a relatively large clearance exists between adjacent flats, however, say of the order of .008″ or more, the soldered joint may be short, for either of two possible reasons. If the depth of dip is small, the solder in the joint will be of like or only slightly greater depth, as there will have been little capillary rise owing to the large clearance. If there has been a deep dip, with adequate superheat, then the solder will fall back out of the gap, leaving again only a short bond. Either of these may give rise to cracks extensive enough to allow serious loss of oil.

Several diverse techniques have been developed to ensure sound joints. In a closely packed matrix, with little or no clearance between the flats, completely sound joints can be obtained by ensuring that the flux has first penetrated all parts of the joints, when the solder will follow it. Now to enable the flux to do this, it must be assisted by having the tube ends thoroughly clean. This entails removing their protective film of oil by degreasing, and any residual oxide film from the end-annealing by pickling. Heating the matrix and the flux to a temperature of about 80° C., also helps the flux to penetrate all the joints by capillary attraction.

Another scheme, which by providing a definite clearance between the flats allows a little more latitude in the foregoing operations, has been developed by Mr. C. S. Steadman, of Marston-Excelsior Ltd. Here the hexagon ends of the tube are stepped. The full-sized hexagon extends from the ends for an eighth of an inch, the remainder being stepped down .004″ across the flats. When the tubes are nested together there will be a space of .004″, which is not too great to hold the solder, and experience to date has shown that a much higher percentage of sound joints can be ob-

tained, when a given technique is employed.

Another method, evolved by Mr. J. Belaieff, of Delaney-Gallay, Ltd., is illustrated in Fig. 42. The assembly jig is gated. When first dipped into solder, the side of the jig is opened, allowing solder to penetrate all the joints. While still in the bath the gate is closed, bringing the block to size and expelling any surplus solder.

Evidence has recently come to hand which accounts for the greater prevalence of matrix face leaks in certain designs of oil cooler. These usually have shallow passes, about 8 tubes deep, yet the distance between the baffles (and hence the width of the dipping jig), has exceeded the nominal dimensions over the tubes by the same amount as in passes of more than twice the depth. There has thus been a greater clearance between the flats, which has been unable to hold the solder.

(*iii*) *Joint between Block and Casing.*

Two types must be considered, (*a*) those, in the majority, where the blocks are dipped in jigs, inserted into the casing, and sealed by hand soldering, and (*b*) those where the tubes are assembled in the casing and dipped with it.

In the former, much of the trouble which has been experienced can be traced to two causes: (*a*) inadequate filling of the matrix so that large gaps exist around the periphery, as shown in Fig. 43, which is a section through the hexagons of a British oil cooler, and (*b*) the design of casing, which makes it difficult for the operative to apply heat so as to allow the solder to penetrate the joint. Fig. 35 illustrates this point well. There is a poor fit, with a large gap. If the operator applied more heat the solder would run through the joint, but in any case, with the jacket construction shown, it is most difficult to get the inner casing hot enough for the solder to bond, even if the block fitted properly. A somewhat better joint might have been obtained by soldering it with the matrix face vertical, *i.e.*, with the joint horizontal, so that

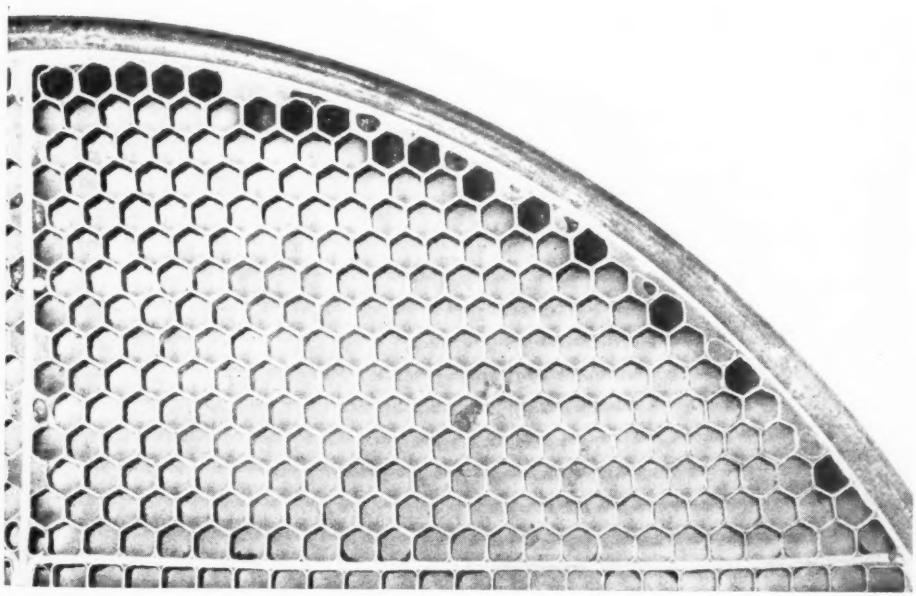


Fig. 43
Section through hexagon joints, showing gaps, partially filled with solder.

the solder would not run away, but inherently this design is bad.

Far better is that in which the jacket space does not extend to the hexagon joints. Fig. 44 shows a section through an American oil cooler, where the dimensional standards ensure that there is always only a single metal thickness around the tube joints. This allows the torch to be applied externally, and by using the heat of the metal to heat the solder, the joint is made under ideal conditions, with results as shown in Fig. 41.

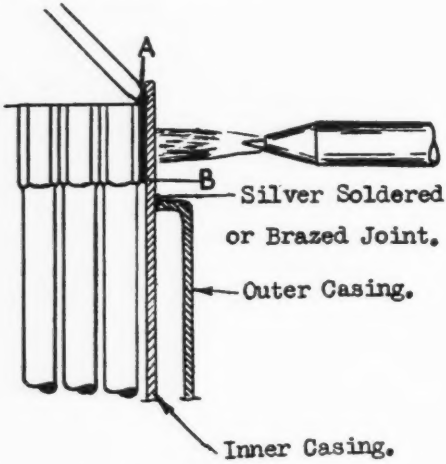


Fig. 44
Section through typical American Oil Cooler joint, showing method of making joint.

Reference has already been made to difficulties which may arise when the matrix is dipped in the casing, particularly when steel casings are used. This method of construction is most attractive from the production aspect, but in order to make it satisfactory, several detrimental factors will have to be eliminated.

The trouble due to differential contraction can be overcome by using only materials of similar thermal characteristics and by ensuring that the design and shape of the casing do not cause unequal expansions. Equally as important as in the hand-soldered case is the fit of the block in the casing. If there are gaps as shown in Fig. 43, the solder cannot rise by capillary attraction, and it will be necessary to finish the joint by hand, with all the difficulties already mentioned.

Fig. 45 shows one method of ensuring that there are no gaps around the periphery, as practised by the Harrison Radiator Division of the General Motors Corporation. The block is assembled with slightly more tubes than the space allows, the jig being left open to permit this. Closing the jig squeezes the tubes together and gives a whole periphery.

In order to fill the block completely, several tube end shapes are used, as shown in Fig. 46. The adoption of a greater number of shapes in this country is recommended.

(*iv*) *Joints Between Block and Baffle.*

Where these joints are made simultaneously with the dipping of the block they are usually sound, as a good fit is obtained by the use of pentagon tubes or corrugated baffles, and the gauge of the baffles is small. Joints made by hand, as in American oval coolers and the majority of British coolers, however, may present a problem. If the baffle comes flush with the surface of the matrix, it is difficult to apply sufficient heat to it, and only its edge is heated so that scanty joints result. A retrograde practice has been to shorten the baffle, so that a gap exists between tube rows of adjacent blocks. This makes it more difficult still to heat the baffle, and the final joint is too often merely a bridge of solder between the tubes, as shown in Fig. 38.

The solution is to allow the baffle to extend beyond the matrix face by an eighth of an inch or more, so that heat can be applied to it by a pencil flame, feeding the solder in from the opposite side, by melting it indirectly by the heated metal, as shown in Fig. 47. This practice cannot be too strongly recommended, in place of the common one of melting the solder in the flame and allowing it to fall on to cool metal, presenting the facade of a sound joint.

(*v*) *Inspection.*

Inspection of the joints of a completed oil cooler is difficult, without sectioning the

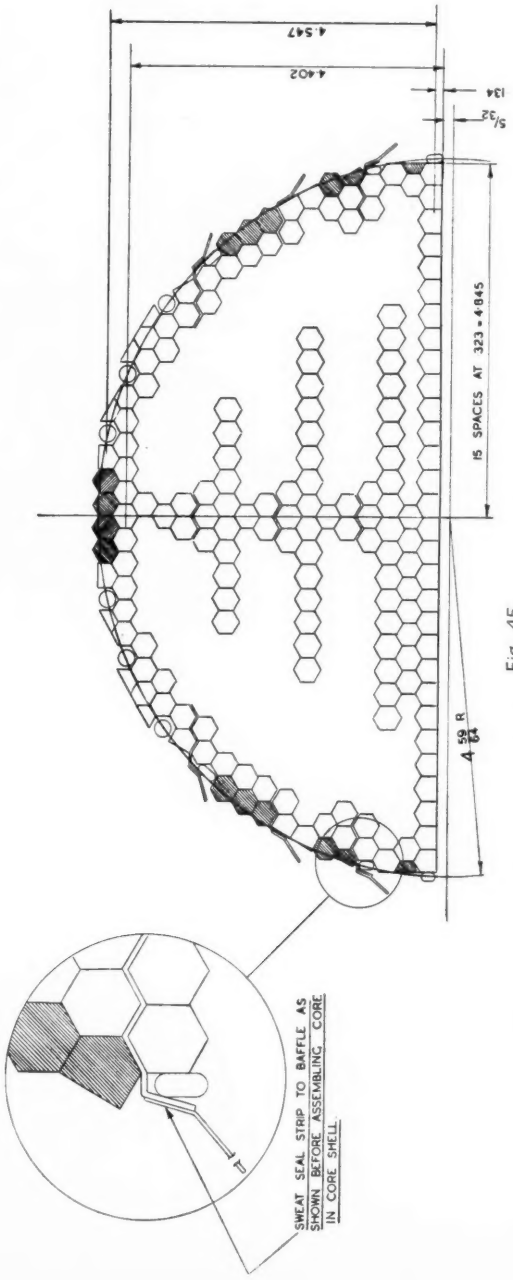


Fig. 45
Sketch showing method used by Harrison Radiator Division, General Motors Corp. to obtain filled periphery by compressing core

cooler. Non-destructive methods of exploring soldered joints which are accessible only from one side are under investigation, and here there is a fruitful field for development.

In the meantime, it is felt that far more could be done by control and inspection of components and stages. The skilled craftsman has in this case been more of a drawback than an asset. He has been prone to accept badly sized parts, priding himself on his skill to make do, without appreciating the duty which the joints have to perform. Then, too, his sturdy independence has made him a poor subject for inspection, and the tendency to leave him many operations to carry out, without breaking them down into stages at which the work could be inspected, has not helped matters.

There is no reason at all why this work should not be planned on modern lines and broken down into suitable stages, as has already been done in the manufacture of automobile accessories where soldering is largely employed.

There is, too, much scope for modern methods such as furnace brazing, for casings, and soldering by induction heating. In view of other developments which are dealt with later, however, it is probable that it is too late for these to be of use.

One valuable contribution has been made by Mr. J. W. Lawrence, who has under development the use of fluorescent liquids applied internally to the components. Ultra-violet light played on to them reveals leaks which cannot readily be detected by other means.

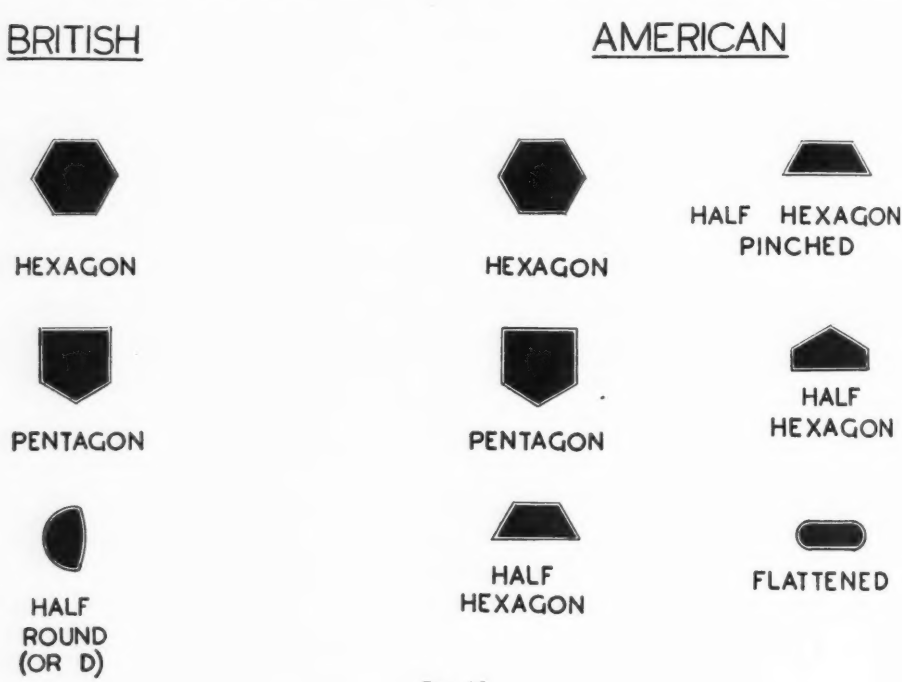


Fig. 46
Standard Shapes of Tube End commonly in use.

(*vi*) *Testing.*

That the practices just mentioned have not been remedied earlier has been due to some extent to the relatively easy production

acceptance tests which have until recently been in vogue. More stringent tests have been under consideration for some time past, but the production situation during the war made it difficult to enforce them earlier.

The existing tests consist of the application of air pressure 50% in excess of the cooler relief valve setting, usually 70 p.s.i., with the cooler immersed in water at 40° C., then immersion in glycerine at 130° C., without pressure, followed by a repetition of the first test. This has been shown to be incapable of detecting every unsound joint, and a single application of the pressure test to units just received from the makers has on occasion rejected a high percentage of them.

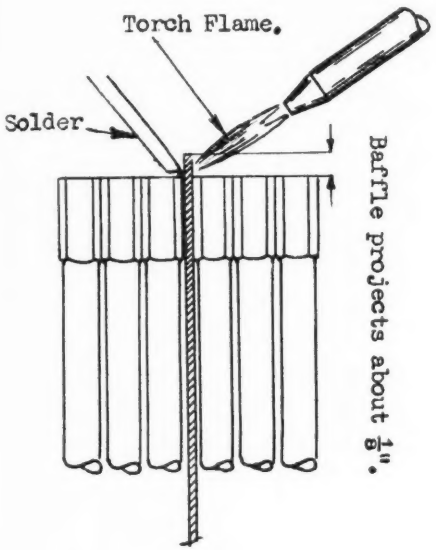


Fig. 47

Sketch showing value of projecting baffles.

Combining the pressure and temperature tests into one has already produced a considerable improvement. Thorough drying before pressure testing, which probably occurred during transit in the case just cited, is also most important, as the presence of even a small amount of moisture inside the cooler will nullify the air-under-water pressure test, by sealing small leaks and by invisible water rather than air being expelled into the test tank

The new M.A.P. test specification, which is just being introduced, aims at combining temperature and pressure, and by repeating some of the tests, alternating hot and cold tests, and ensuring that the cooler is thoroughly dry internally, it has become far more searching than the previous one. Details of the new test procedure are given below:—

(*i*) Immerse in water at 40° C., with an internal air pressure of 70 p.s.i. (or higher) for 5 minutes.
(*ii*) Immerse in a bath of glycerine at 130° C., for 5 minutes.
(*iii*) Ensure that the cooler is thoroughly dry internally.
(*iv*) Repeat test (*i*).
(*v*) Apply a pressure of 70 p.s.i. (or a hydraulic pressure of 70 p.s.i. (or higher) at 130° C. for 5 minutes.
(*vi*) Apply the same pressure as in test (*v*), hydraulically, for 5 minutes at room temperature.
(*vii*) Repeat test (*ii*).
(*viii*) Repeat test (*i*), the cooler being thoroughly dry internally.
(*ix*) Store the unit for a period of not less than 8 hours.
(*x*) Repeat test (*vi*).
(*xi*) Repeat test (*ii*).
(*xii*) Repeat test (*i*).

The production acceptance tests employed in America comprise a static pressure test of 100 p.s.i., cold, with a percentage (0.5%) of production components subjected to a pressure cycle test. This consists of applying air pressure, six cycles per minute, varying from zero to 60 p.s.i., while the cooler is immersed in oil at 150° C. Any three consecutive samples must average 15,000 cycles without leakage, no figure above 20,000 cycles being counted.

From tests which are still incomplete there are indications that a high proportion of British oil coolers of current manufacture will be rejected by the new M.A.P. test, and

that few, if any, will survive a sufficient number of cycles of the American test to rate as minimum acceptable production samples. On the other hand, there is every indication to support the view that a considerable improvement is possible, without change of design.

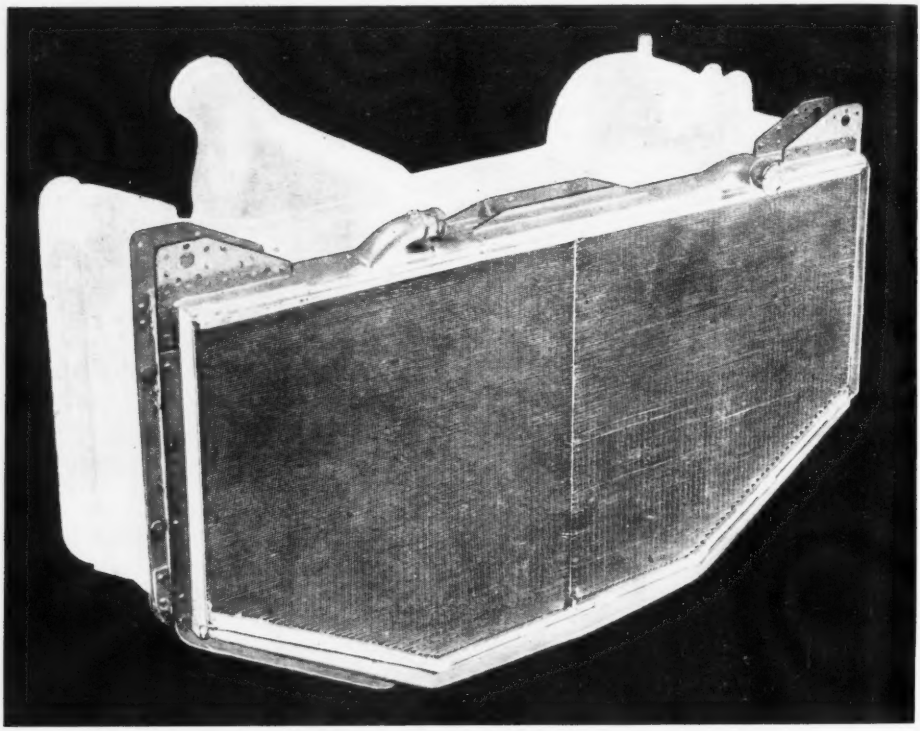


Fig. 48
Morris Secondary-Surface " Series " Oil Cooler for Merlin-Lancaster Power Plant.

8. NEW DEVELOPMENTS.

Although the honeycomb oil cooler is so universally employed, we have seen that it possesses a number of defects, the obviation of which involves a greater or less measure of weight, complication, cost, or drag. Several developments are reaching fruition which avoid some of these drawbacks.

8.1. *Secondary Surface Series Oil Coolers.*

Pioneered by Dr. E. W. Still, of Rolls-Royce, and Morris Motors Ltd. (Radiators Branch), the latest Merlin installations use a secondary surface oil cooler mounted directly behind the coolant radiator, as illustrated in Fig. 48.

This oil cooler is made in the same way as a secondary surface radiator of fin-and-tube type, the only difference lying in the provision of oil tube spacers which are corrugated in a special manner to promote turbulence and increase heat transfer. It has a far greater total cooling surface than the honeycomb design, and contains less oil.

The design has far fewer joints than the honeycomb type of equivalent capacity, and experience has shown it to be much more reliable. Being mounted behind the radiator, the air entering it is warmed, so that over-

cooling and excessive congealing are reduced to a minimum. Combined with the radiator in this way, it is only possible to compare the efficiencies of the two systems together. The radiator has three rows of water tubes instead of four, and is able to have a greater frontal area by absorbing the space normally occupied by the honeycomb oil cooler, a greater overall efficiency resulting.

8.2. *Galley Mixed Matrix Oil Cooler/Radiator.*

In Fig. 49 is shown a combined oil cooler/radiator in the Tempest installation, and in Fig. 50 a section of one cooling element. The component is of secondary surface type, but the liquid tubes consist of narrow containers closed at one end. These are divided longitudinally, and provided with

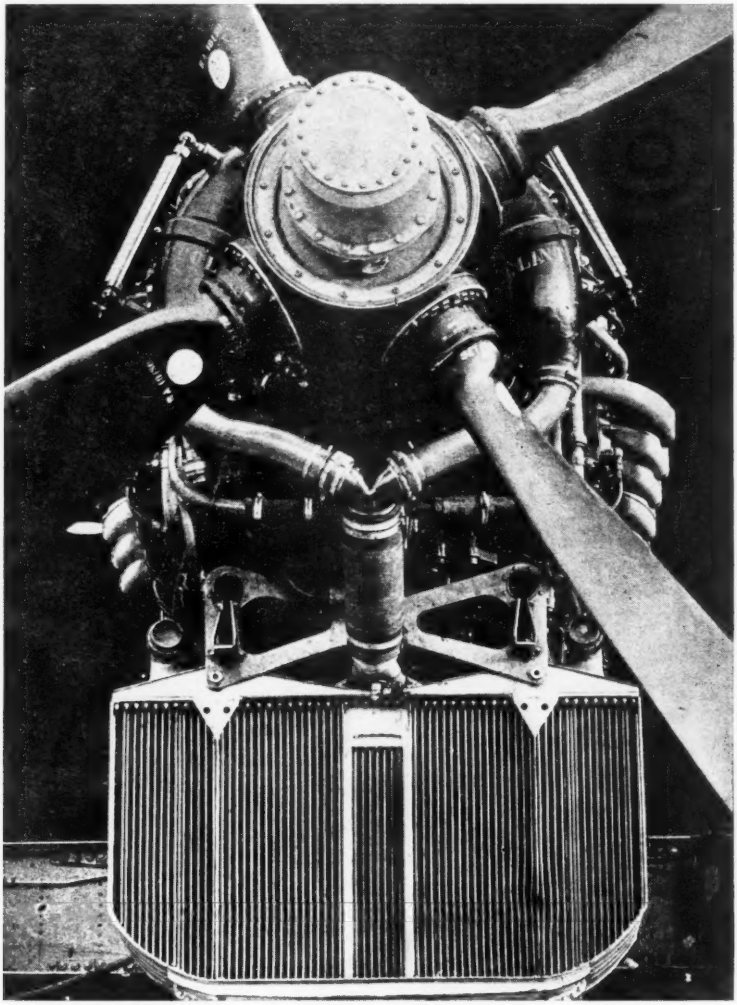


Fig. 49
Delaney-Gallay Mixed Matrix Oil Cooler/Radiator in Sabre-Typhoon.

perforated stiffeners, so that the liquid enters one side at one end, passes U-wise down that side and up the other, and out at the same end. By fitting these "tubes" alternately opposite ways, it is possible to have an oil tank at one end, and a coolant tank at the other. The number of tanks, and consequently the number of tube plate/tube joints is halved.

This type is manufactured by forming the "tubes" and stiffeners from tinned copper sheet, assembling them in a jig, with tinned corrugated strip between them to form the secondary cooling surface, clamping the jig,

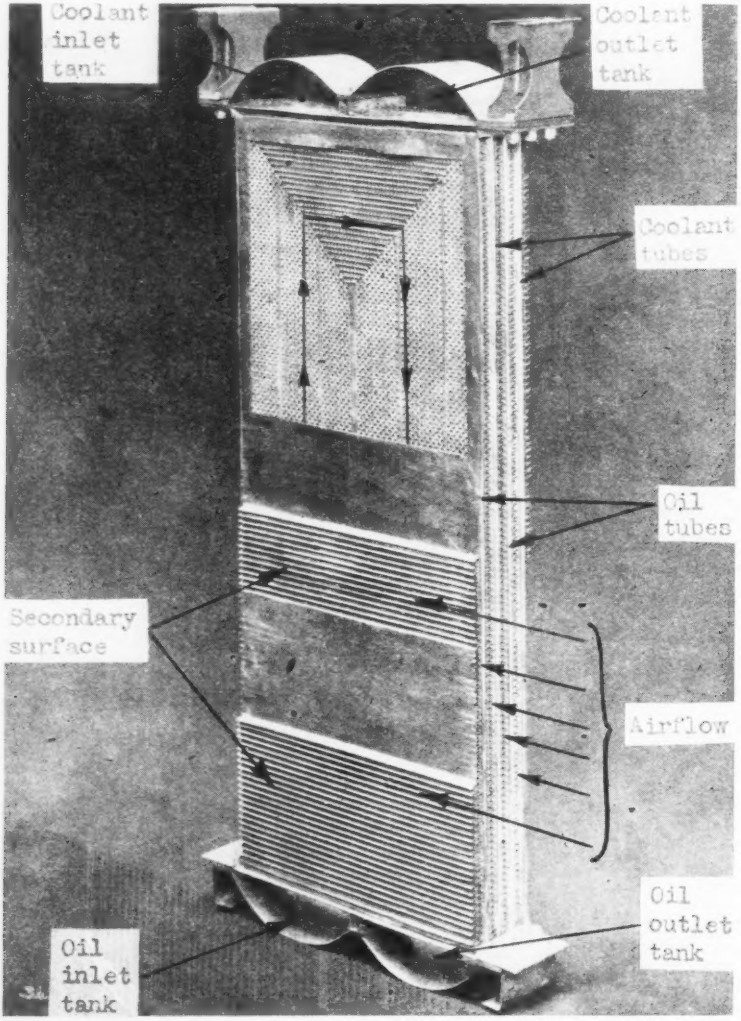


Fig. 50
Section through Delaney-Gallay Mixed Matrix

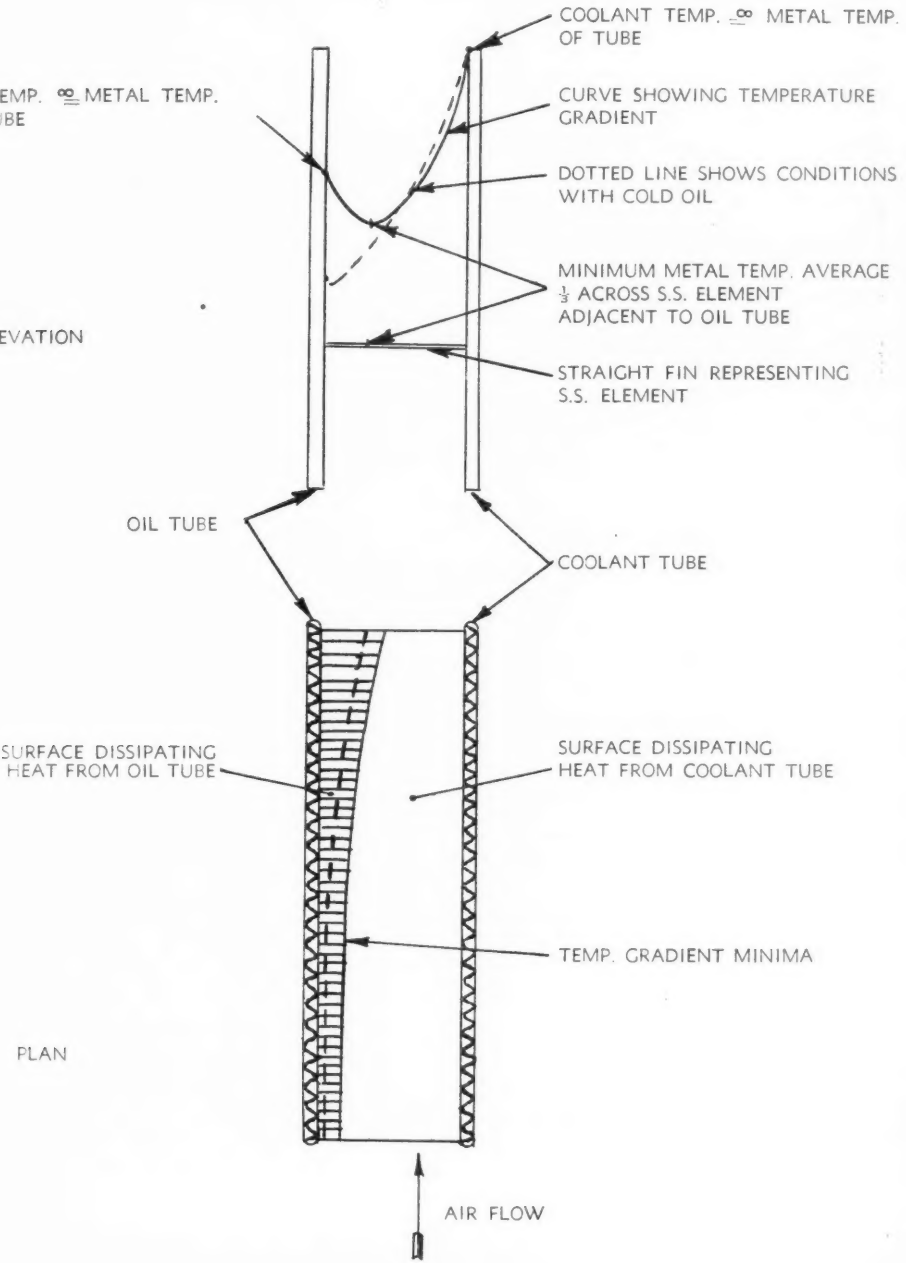

COOLANT TEMP. ≏ METAL TEMP. OF TUBE
OIL TEMP. ≏ METAL TEMP. OF TUBE
CURVE SHOWING TEMPERATURE GRADIENT
DOTTED LINE SHOWS CONDITIONS WITH COLD OIL
MINIMUM METAL TEMP. AVERAGE ⅓ ACROSS S.S. ELEMENT ADJACENT TO OIL TUBE
ELEVATION
STRAIGHT FIN REPRESENTING S.S. ELEMENT
OIL TUBE
COOLANT TUBE
SURFACE DISSIPATING HEAT FROM OIL TUBE
SURFACE DISSIPATING HEAT FROM COOLANT TUBE
TEMP. GRADIENT MINIMA
PLAN
AIR FLOW

Fig. 51
Diagram illustrating functioning of mixed matrix.

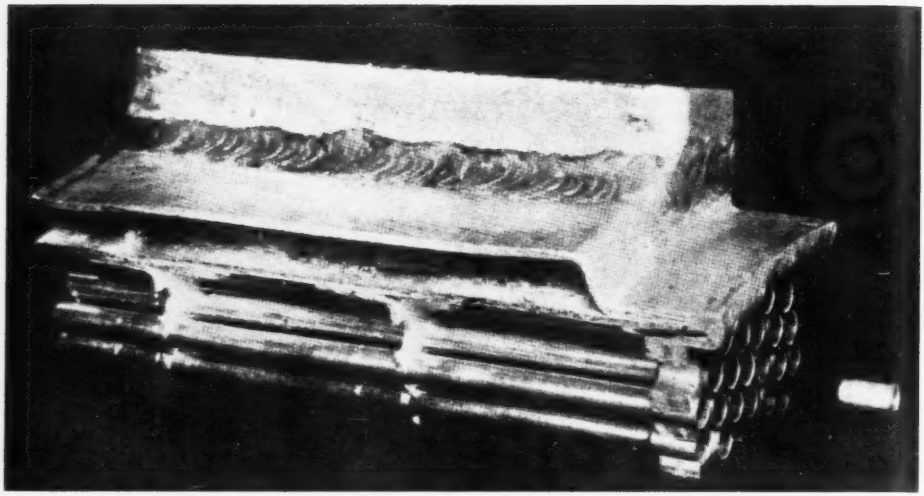


Fig. 52
Section of Fedders Cross Flow 12" Aluminium Oil Cooler showing tube/Header plate joints.

and baking the whole in a furnace, so producing a rigid block to which suitably divided tanks are fitted.

Important advantages are claimed for the design. For the same air flow this mixed matrix dissipates more heat than any other type—secondary surface series, or secondary surface radiator with honeycomb oil cooler, so that for a given installation there may be a weight saving of as much as 18%.

Then from the point of view of oil cooling, a measure of automatic compensation is obtained. To explain this, Fig. 51 has been prepared. Since each cooling fin is working for both a coolant tube and an oil tube, and the mean temperature of the coolant will exceed that of the oil, there will be a temperature gradient across the fin as shown. Under conditions of low oil inlet temperature, this gradient will change to the form shown dotted. In other words, the area of fin which is dissipating heat from the oil tube will be reduced. As the air warms up in its passage through the radiator, so will the proportion of fin cooling the oil increase.

The coolant tube thus takes maximum advantage of the secondary surface when the air is at a low temperature, and the oil tube makes fuller use of the portion swept by the warmer air.

The design is quite flexible, and any desired proportion of the whole can be used for oil. It also lends itself to annular designs, where the tubes remain straight, being disposed radially, and also make best use of the air by having expanding air passages between the tubes.

8.3. *Aluminium Honeycomb Oil Coolers.*

Several firms in America are now producing honeycomb oil coolers made entirely of aluminium alloy, with a weight saving of about 50%.

Starting with hexagon-ended tubes dipped in an aluminium brazing alloy, this was dropped in favour of an all-mechanical joint. Plain parallel tubes are threaded through header plates having holes drilled at centre distances which give the same oil passages as in the soft-soldered copper design. Small ferrules are inserted in the tubes, which are then expanded by a rotating tube expander, forming a seal which will pass the rigorous American test schedule. Fig. 52 shows a sec-

tion of one such cooler, made by the Fedders Manufacturing Co., and Fig. 53 gives a view of the complete unit.

Conservatively, work began with tubes thicker than the normal copper ones, but with the confidence now gained there is a trend towards tubes .006″ thick.

It is claimed that any defective or damaged tubes can be replaced in the field, simply by extracting the bad tube, inserting a new one, and making a seal with a rotating hand tool.

The chief attractions of this design are its inherent strength at the highest oil temperatures likely to be encountered; its dependence upon normal shop practice instead of coppersmithing; and its lightness.

Since it is not possible to use half tubes to fill the irregular spaces at the ends of the rows, it is necessary to prevent excessive by-passing of hot oil around the periphery of the

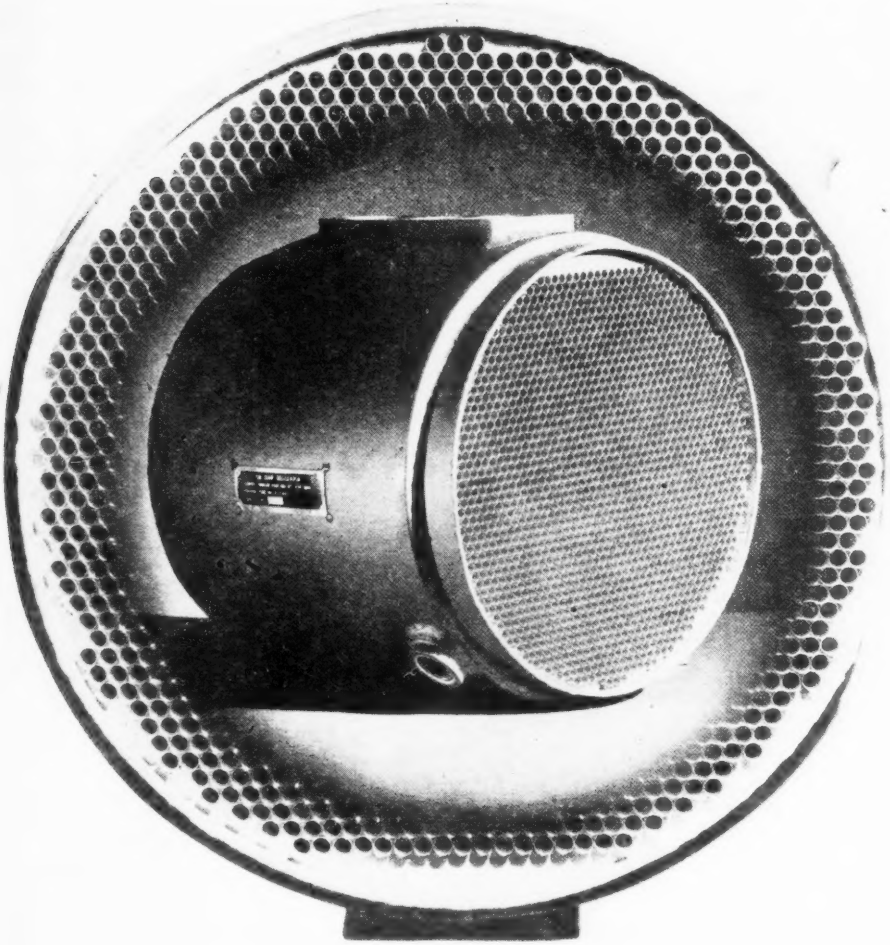


Fig. 53
12″ Diameter Fedders Oil Cooler with Aluminium alloy tubes.

block by packing the spaces with aluminium foil.

Fig. 54 shows one such oil cooler by Airesearch. This is of the standardized oval shape, with a frontal area of 1.91 sq. ft., and a unit heat rejection well in excess of specification requirements. It weighs only 35 lbs.

8.4. *Jacketless Honeycomb Oil Coolers.*

American oil coolers are standardized to such an extent as regards size, performance, and reliability, that price and weight are the only avenues left open for competition. In order to meet the challenge of the aluminium cooler, United Aircraft Products Corp. have evolved a range in which weight saving has been the main object, while retaining orthodox methods of manufacture.

This has been achieved by doing away with a jacket, saving both metal weight and oil weight, and using tubes .004″/.0045″ thick, in brass. A pictorial representation of this oil cooler is shown in Fig. 55. A weight saving of about 15% is possible.

Despite the lack of a warming jacket, this design decongeals satisfactorily without control of the airflow, mainly by reason of the diffusion of hot oil from the central tubes, and through the diffusion passages formed by the use of a large number of dummy tubes.

The outer casing is stainless steel. The

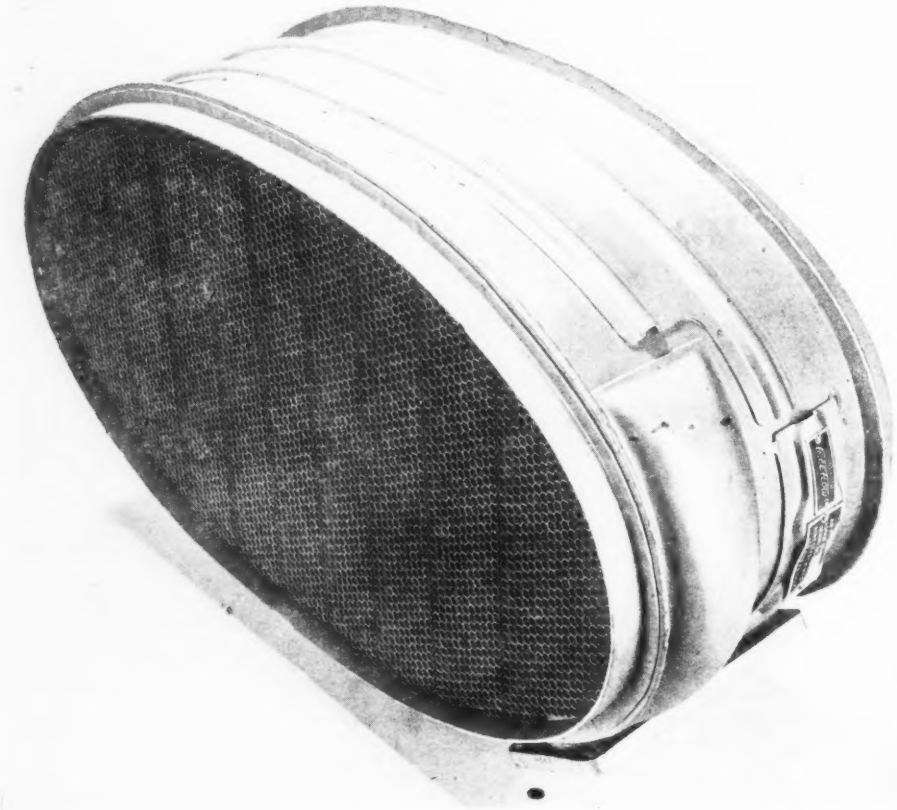


Fig. 54
Oval Aluminium Oil Cooler by Airesearch

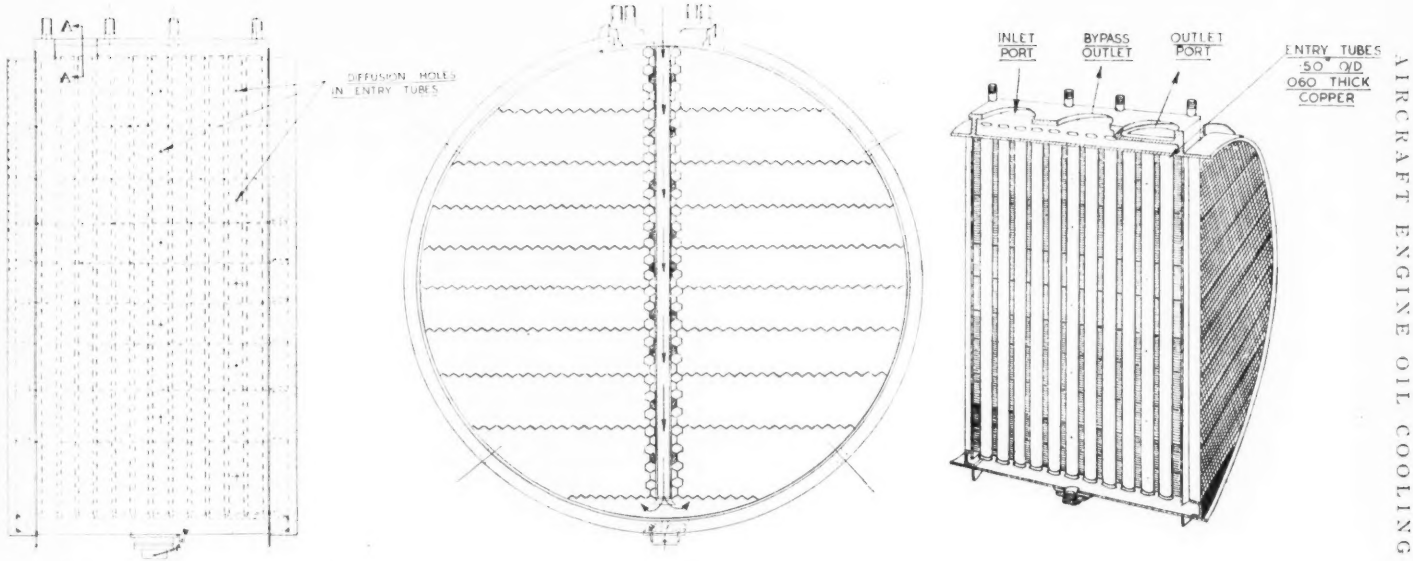
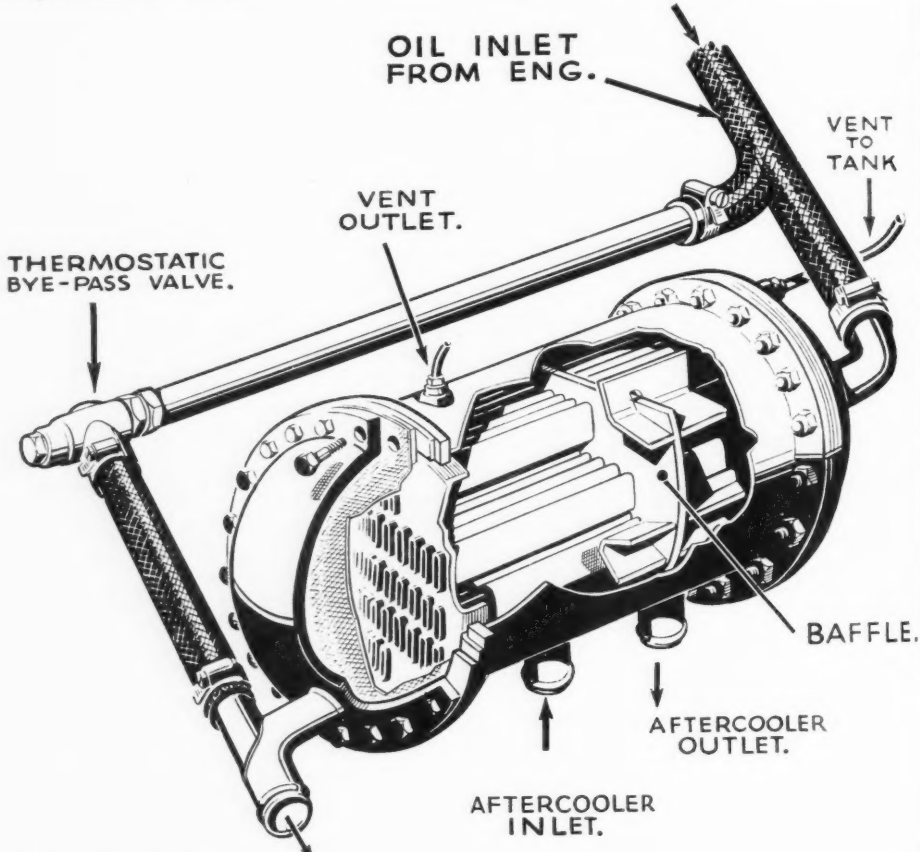


Fig. 55
Details of UAP Jacketless Oil Cooler

central tubes are quite thick, and it is claimed that they eliminate the need for a surge protection valve. When the oil cooler is frozen, the resistance of the oil in these tubes gives full protection to the matrix.

8.5. *Heat Exchangers.*

The latest Mustang has an oil cooler following the lines shown in Fig. 56, in which coolant from the supercharger aftercooler circuit is the cooling medium. This cooler is



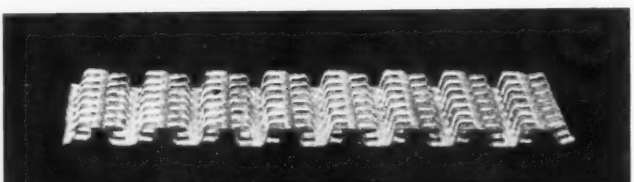


PHOTO SHOWING SECTION OF TURBULATOR

Fig. 56

Diagrammatic Sketch of Harrison Heat Exchanger used on P.51H Mustang

small and compact, and is installed behind the engine just forward of the oil tank. Oil lines are thus kept to a minimum, and in fact there are none aft of the firewall. Coolant passes from the oil cooler to a radiator element which is part of and in front of the main radiator unit. From this it passes through the after-cooler, and back to the oil cooler.

A full measure of control is possible by inserting a thermostatic by-pass control in the coolant line.

8.6. *Wet Sump Systems.*

Though belonging more properly to the study of the oil systems as a whole, mention must be made of the advantages of a reversion to a wet sump, as they have a bearing on the trend of oil cooler development.

Fig. 57 (Ref. 23) shows one system which has been proposed. The hopper is removed from the main oil tank, which now functions merely as a header or make-up tank, and is transferred to the engine as a wet sump. A

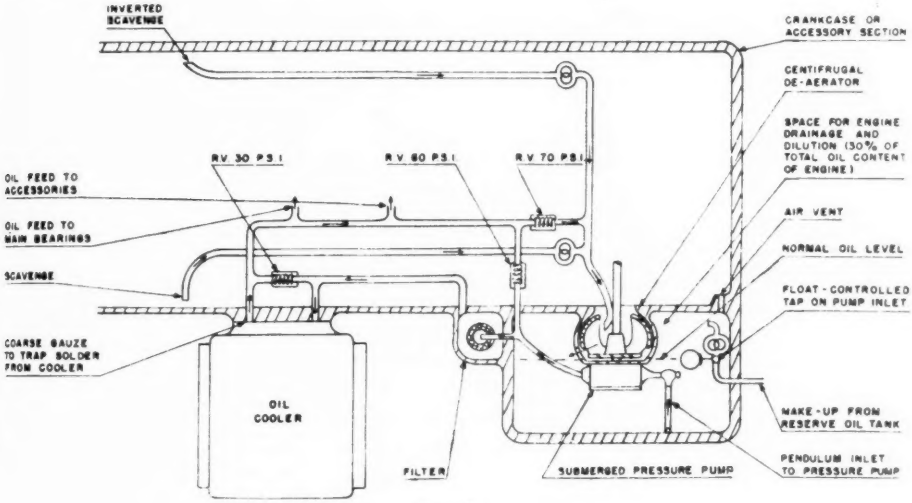


Fig. 57
Diagram of Wet Sump Oil System for Aeroplane Engines.

It will be remembered that a scheme similar to this was fitted to the Wright Tornado engine about 18 years ago.

Another heat exchanger has been used on some of the German installations of Junkers engines. In this case the oil has flowed over a finned tube through which coolant has been passed. Unfortunately no performance figures of this, and no oil figures for the engine, are available.

An appreciable saving in weight and drag has resulted in the Mustang case, as compared with the honeycomb oil cooler/secondary surface radiator combination, due to some extent to the elimination of the long oil pipe lines of the earlier installation.

submerged pressure pump in this sump draws oil from its base through a pendulum pipe, and forces it through the oil cooler by way of a filter, and so to the engine. Scavenge pumps return the oil to the wet sump through a centrifugal de-aerator, and the level of oil in the sump is maintained by a small pump in a communication with the oil tank, controlled by a float-operated valve.

With this system there is only one oil pipe between the engine installation and the oil tank. Vulnerability is greatly reduced, there is reduced fire risk as only the contents of the sump can reach a fire in the nacelle, since the rate of replenishment from the main tank is low, and the reserve oil need not be changed

in the event of engine failure. In addition, there will be far less aeration, since there is no back pressure on the scavenge pump, while there will be quicker warm up and a considerable simplification of the system to suit oil dilution for cold starting.

From the point of view of the oil cooler the advantages are (*a*) that as the system is self-contained with the engine one oil system only need be developed, and not one for each

bilities here, as, too, does the furnace brazed heat exchanger.

An investigation made at the R.A.E. (Ref. 24) into the requirements of oil systems for rapidly-detachable power plants has led to a somewhat similar system. Here, however, by having a separate pump for the oil cooler and another to feed the engine, increased pressures on the cooler are avoided.

Fig. 58 (Ref. 24) shows the arrangement of

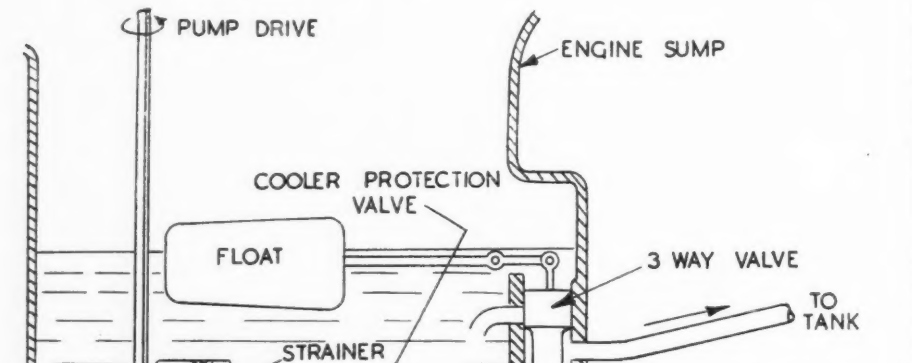


Fig. 58 *(Crown copyright reserved.)*

Diagram of three-way Float Valve Power Plant Oil System.

different installation; (*b*) that warm-up on starting will be quicker, reducing the duty on the oil cooler; and (*c*) that with the elimination of long pipe lines, especially between scavenge pump and oil cooler as in existing installations, there will be less risk of overcooling of the oil.

The one drawback lies in the high pressures to which the oil cooler, now in the pressure circuit to the engine, will be subjected. The secondary surface construction offers possi-

the sump and pumps. A three-way float valve maintains the oil level in the sump by replenishment from the main oil tank. An independent pump circulates the oil through the cooler, and back through a de-aerating junction box to the main pressure pump for delivery to the engine. Should the level of oil in the sump rise, the cooler pump would return oil to the tank via the three-way valve.

By providing a relief valve in the line between pump and cooler, to divert the oil to

the sump when the cooler back pressure was high, the need for a cooler by-pass would be obviated.

The use of an independent cooler pump gives an advantage over the first scheme which is of particular value to British air-cooled engines, in that the rate of circulation through the cooler can be as high as the oil cooler designer requires, regardless of the engine circulation.

Other advantages are much as outlined in the first scheme, two not already mentioned being a constant supply of hot oil for the air-screw feathering pump, and a small but definite head on the inlet side of the main pressure pump, of value when operating at high altitudes.

8.7. *Future Trends.*

With the rapid development of the i/c turbine, oil cooling trends during the next five years or so are going to be divergent. On the one hand, in the field of civil transport, we are likely to see engines of around 4,000/5,000 H.P. in single units, while in high-speed fighter aircraft the turbine unit will probably replace the reciprocating engine entirely.

The large reciprocating engine will be more totally enclosed than at present, with a reduction in incidental cooling from the crankcase, etc. At the same time, with increasing specific outputs as the competitive pressure from the turbine makes itself felt, there will in all probability be greater use made of piston and bearing cooling by the oil. Conservatively, therefore, the heat to be dissipated from the oil of a 4,000 H.P. power unit will be of the order of 400 H.P.

This is a startling thought, not that the percentage figure is any greater than now, but because of the weight and drag involved in an aircraft which must pay its way commercially.

Here then is much scope for development. In order to make the cooler smaller we must have much higher speeds of flow, involving far higher pressures than heretofore, and requiring in turn stronger methods of construction, mechanical or brazed. The merits of cooling by forcing the oil through one long passage instead of through a multitude of small passages in parallel, need to be investigated, and withal there should be drawn up a balance between the power required to pump the oil, and that expended in weight drag and cooling drag.

Cooling drag needs to be studied far more fundamentally, and in large aircraft there is an excellent case for the canalizing of all the air services through a common set of ducts. Thus air for engine consumption, engine cooling, oil cooling, and cabin heating should be considered as a whole. In such a layout it should be possible to take warmed air for the oil cooler, and have the latter add its meed to the air for cabin heating.

Returning to the turbine engine, we have seen that present requirements are small, being dictated only by the bearing and gear cooling characteristics of a particular design. As powers increase, there is no indication of an increase in the percentage heat-to-oil, and it is fairly safe to assume that for a long time ahead we are not likely to require coolers any larger than those at present in use. Apart from any fundamental advances which may be made in oil cooler design, therefore, it remains to consider in what directions lie the most desirable improvements.

Taking the cylindrical or oval cooler as representing the simplest and soundest type, the use of aluminium alloy, with mechanical joints, holds much promise. From the cooling aspect, the author has long felt that a fuller knowledge of the oil flow conditions inside a honeycomb oil cooler would be of great value, and a programme of work, using passes constructed of perspex, to investigate the effect of pass shape and proportions, baffle port shape, size, and disposition, should pay dividends.

It is only within the past three years or so that cold wind tunnels have been available for full performance testing of oil coolers, and the number is still too small to cope with existing development and research programmes. This situation is being rapidly remedied, and it is hoped that soon every manufacturer will

have his own cold tunnel. In parallel with this work there is need for full-scale flight research at altitude, to measure all the quantities involved and to confirm present assumptions.

As the basic causes of the technical difficulties of oil cooling lie in the properties of existing types of lubricant, there is naturally a keen desire to see an oil developed which will not become excessively viscous at low temperatures. There are at the moment of writing well-substantiated hopes that such an oil will be available in the not too distant future.

With all this, and whatever direction taken by future developments, there is now and always will be a vital need for a better realization in the manufacturing industry of the importance of the oil cooler to the safety of the aircraft, and a more widespread knowledge of the duty which it has to perform. This should inevitably lead to an appreciation of the value of sound detail design and, above all, of the value and absolute necessity of production engineering.

ACKNOWLEDGMENTS.

My grateful thanks are due to all those referred to in the text, and in particular to

(1) My colleagues in Roll-Royce Ltd. and at M.A.P. for much helpful discussion.

(2) The members of the aircraft heat exchanger industry, for information supplied, and for having patiently borne much well-intentioned criticism.

(3) Numerous friends in America, for so freely revealing their manufacturing processes.

It should be added that the views and opinions expressed in the Paper are entirely my own, and do not necessarily represent those of my Company nor of M.A.P.

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DISCUSSION

MR. F. ROWARTH (M.A.P., Fellow), after congratulating Mr. Nixon most heartily on his excellent and comprehensive paper, said it was so often the case that when a really conscientious review of that kind was made it showed that we tended to go round in circles. Frequently the early ideas were very sound, but were dropped through lack of appreciation of the problem or inadequate development; but later on we came round to them again, because the alternatives had definitely come to the end of their period of utility.

The instance in the case under consideration was the mechanical construction of oil coolers. Ten or fifteen years ago we had had coolers constructed by the means shown in Fig. 10 of the paper; but, because of inadequate development, they were dropped. Now we had come back to them, and in some quarters they were hailed as something new. Mr. Nixon's paper indicated clearly that they were not.

The problem of cooling oil on a pre-war engine was comparatively simple; but developments during the war—both directly, in the new engines themselves, and indirectly, in the higher aircraft performances obtained, the higher altitudes reached and the universal sphere of operation—had taken the problem of oil cooler design from the stage of an elementary to a quite complicated process.

Unfortunately, only those really intimately concerned with the job had appreciated that to any real extent. To the ordinary technician in the aircraft business the oil cooler tended to be taken for granted, and even some of the engine manufacturers had been extremely unsympathetic to the problem.

Table I in Mr. Nixon's paper showed the increases in heat-to-oil that had taken place ; but that, of course, was only half the problem. The greater range of operating temperatures had also had very considerable effect.

Although his remarks so far had been sympathetic to the difficulties of the oil cooler designers, and he was aware that the manufacturing side had had its own peculiar troubles, he did not think that sufficient technical brains had been available to the industry to keep pace with the ever-increasing problem. In that connection he suggested that Mr. Nixon had extended too far our national privilege of understatement when he had said that "The oil system has been by no means the most reliable and trouble-free section of the aircraft during the war."

The magnitude and ramifications of the oil cooling problem had not been appreciated by the senior members of the aircraft world. Consequently all aspects of development had lagged sadly behind, with the result that the oil cooler actually had been one of the most unreliable items of equipment on aircraft during the later years of the war, with very few exceptions.

During the past 18 months, however, there had been considerable activity to try to improve that condition of affairs, and it was only fair to state that considerable improvements had been made. It was not unreasonable to add that Mr. Nixon had

played "a not unimportant part in those activities!"

As an indication of the results obtained, he recalled that some 12 months ago the Maintenance Units were rejecting an appallingly high percentage of new coolers; but notification had just been received that they were no longer going to test the coolers on arrival because the percentage of rejects had fallen to practically nothing.

The next stage, having achieved some measure of reliability on present practices, was to improve the practices. It was absurd that in 1944, with inlet oil temperatures of the order of 130° C., we were still using soft solder, which, as shown by Fig. 32 of the paper, had lost half its tensile strength at that temperature. We must get away from soft solder— and that very quickly.

As Mr. Nixon had shown, in the United States, where admittedly technical manpower had been plentiful, they had adopted almost universally the light alloy oil cooler, and had now achieved a light alloy radiator. Those products offered dividends in weight which were most attractive to aircraft designers and operators, and although the turbine was coming along fast, there would be for many years aircraft types for civil use where a saving of 50% on the cooling equipment would represent a very handsome increase in payload.

In a concluding expression of thanks to Mr. Nixon for his valuable historical and technical review of the oil cooling art, he also thanked him (and Messrs. Rolls-Royce for having allowed him to do it) for the great assistance which his energy and constructive criticism had been to the M.A.P. for the past 15 months.

MR. J. BELAIEFF (Delaney Gallay, Ltd.) referred to newly developed oil coolers for liquid-cooled engines and thought it would be useful to stress certain differences between the Secondary Surface oil cooler referred to as "series" type and the "mixed matrix." The first, being positioned behind the radiator, was without doubt very well protected against any coring tendency under temperate or Arctic conditions. In the Tropics however, this design could be handicapped on installations where the mean oil was lower than the mean coolant temperature. The Mixed Matrix, on the other hand, was constructed in such a manner that advantage was taken of the full temperature difference between the mean oil and ambient air temperatures.

DR. E. W. STILL (Rolls-Royce, Ltd., Associate Fellow), after complimenting Mr. Nixon on his very courageous summary of the position, recalled that those concerned with oil coolers in the early days of the war had suffered severe headaches; and he emphasised Mr. Rowarth's remarks.

Speaking generally as one engaged on the development side of the Aircraft Industry, he said that the Industry might have been better prepared to face the war than, in fact, it was if there had been better development facilities. For instance, it was not until the war had started, when they had had to go outside the then existing aircraft radiator industry for production expansion, that they were able to get what had become known as the secondary surface radiator. That difficulty was due entirely to lack of development facilities in the radiator companies at that time. Once this radiator type was established, the whole industry had had to give way, because the prices of radiators had dropped enormously, and the amount of labour employed in the making of each radiator was reduced. It was only as the result of that change, which had occurred about 1940, that they were able to face up to the tremendous expansion which had occurred in the radiator industry and in the production of cooling units. Production had increased from somewhere about 30 tons per month in 1940 to more than 300 tons per month in the later stages of the war period.

That had had several effects. It had caught the industry on the hop, for they

were completely in the hands of the sheet metal workers for the original production of 30 tons per month. They were paying about 12/- per lb. for honeycomb tubes ; and just before the war they were produced by 15 drawing operations per tube.

That was the background against which they had started the war. He had come into the industry a few years before the war, and he had been very shaken when he had found that aircraft were relying on honeycomb tube radiators or oil coolers, bearing in mind that they involved thousands of soldered joints. On the latest designs, as the result of experience during the war, they had completely eliminated honeycomb tube radiators and oil coolers. He considered that as a type they were dead, except perhaps for the air-cooled engine.

The secondary surface type of oil cooler placed behind the radiator could withstand very high pressures, and the troublesome matter of coring could be eliminated.

The Americans were still designing their oil coolers "by guess and by gum," in spite of Mr. Nixon's remarks. They would put the coolers into a cold wind tunnel and, if the oil did not flow around a certain portion, they persuaded it around, by providing hot spots, that lost total cooling surface. But oil would sometimes do exactly the opposite from what one expected; it was a most difficult liquid to play with when cold.

Great credit was due to the M.A.P. for having backed the installation of the cold wind tunnels, which were made available to the radiator industry. From the point of view of Rolls-Royce, Ltd., with their types of coolers—they did not experience quite so many troubles as the air-cooled engine people—it could be said that, on the experience gained from cold wind tunnel work, they could design an air cooler which would not core. They could be pretty certain that when a new design of oil cooler went into the tunnel it would be all right.

He did not think the honeycomb oil cooler of the circular type for air-cooled engines was quite so simple; very often there were very big temperature differences, the rise of temperature across the engine was much greater than in the liquid-cooled engine, and the troubles were greater. In some ways he had much sympathy with the radiator and oil cooler manufacturers in their efforts to design oil coolers for air-cooled engines; the problem was not nearly so simple as for the liquid-cooled engines.

Another trouble which had been experienced in the early days of the war was in connection with the soldering up of the tubes around the casing. Unless one could make certain that a tube was completely free when one looked through it, or if it had been soldered up at one end, then he advised soldering it at the other end, because otherwise, if the aircraft in which it was fitted were left in the open, water would get into the tube and freeze, and the tube would split. There had been a lot of trouble with the Halifax oil coolers at one time as the result of a detail such as that.

Emphasizing the importance of designing with a view to quantity production, he said it had been brought home to him very forcibly, in his capacity as a Development Engineer for Rolls-Royce, Ltd., that in the course of development work one was apt to get a technical answer to a problem which did not completely fill the bill. Manufacturers were not all instrument makers, and the designer must bear in mind that the job had to be made on a production basis. In arriving at the initial design the designer should be very careful to ensure that when the job went into production it could be made by unskilled labour. In his view, three-quarters of the trouble that had been experienced in the manufacture of radiators and oil coolers had been due to the designs having been arrived at initially on the basis of the production being in the hands of sheet metal workrs having high qualifications. But that practice had broken down

completely, because of the tremendous expansion that had been necessary. In the future that aspect would have to be watched very carefully if we were to be able to guarantee 100% results.

Commenting that quite a number of people concerned with aircraft engines might have been severely shaken by the revelation in the paper that there were hundreds of different ways in which oil coolers could go wrong, Dr. Still endeavoured to ease their minds by pointing out that we had learned quite a lot during the war. We had learned how to make oil coolers simpler, and he was quite confident that we could go forward in future with the knowledge that we could guarantee our power plants, both in respect of oil coolers and radiators, for transport aircraft without any worry whatever, provided all the results which Mr. Nixon had derived from his work were applied faithfully by the industry. Apparently he was not the only person who thought so, because this country had managed to take from the Americans the orders for the power plants of the DC-4 for Trans-Canada Air Lines.

MR. R. L. LICKLEY (Hawker Aircraft, Ltd., Fellow), speaking as one concerned with aircraft design, said he was very interested to learn how difficult it was to make a good oil cooler and how many of the problems in connection with aircraft operation were due to bad manufacture of oil coolers.

The problem set for the aircraft manufacturer was very difficult, because not only were there engines in which the oil flow varied by $\pm 30\%$ from engine to engine, but one type of engine might have three times the oil flow of another type. To deal with this latter variation he might either have a large oil cooler with low oil velocity, or alternately the quite different problem of a small oil cooler with much higher oil velocity. It was very pleasing that Mr. Nixon had discussed this problem of oil velocity, because the aircraft manufacturers had suffered a lot in the past from the attempt to keep the oil velocity low. In that connection he asked what back pressure was considered permissible in a good modern oil cooler, because one could push up the oil velocity quite well if one were prepared to accept high back pressure.

With regard to the point that, by using a light alloy radiator instead of the present type of brass honeycomb type, considerable saving of weight could be effected, he suggested that some weight saving could be achieved if the amount of solder used were controlled more carefully in manufacture. Further, he asked whether the life of a light alloy radiator was as good as that of a well-designed and carefully-manufactured brass one.

He also reinforced Dr. Still's plea that coolers should be designed with a view to ease of production, because a lot of trouble had arisen in that connection in the past.

Referring to a cooler which Mr. Nixon had illustrated, from which certain tubes were left out with a view to facilitating warming up, he asked whether that arrangement, which enabled the cold oil to get through easily, would not on the other hand reduce considerably the duty at higher temperatures.

Finally, he recalled that one installation which had a combined oil cooler and radiator in a controllable duct, and which was not mentioned by Mr. Nixon, was that fitted to the Hurricane in 1934-35.

MR. L. G. SAVAGE (M.A.P.) felt that as the lecturer and subsequent speakers had dealt so comprehensively with the subject, he could not usefully contribute to the discussion; but, having heard the accusations made by the aircraft and engine manufacturers and the M.A.P., he hoped that the manufacturers of radiators and oil coolers would express their views, particularly in relation to the difficulties with which they were faced.

MR. C. S. STEADMAN (I.C.I., Marston Excelsior Ltd.), as one who had been closely connected with the oil cooling busi-

ness, said that he must to a large extent accept the criticisms that had been made of it; but he felt that he could afford to take up the cudgels on its behalf.

In the first place he pointed out that in the early stages of the war most aircraft constructors and some engine constructors had taken only a negligible interest in oil cooling or oil coolers. The result was that many units were put into large-scale production without having been tested in the real sense in aircraft installations. It might be argued that it was up to the cooler manufacturer to make sure that his products would function 100 per cent. without such pre-testing; it would be very nice to be able to do that, and they were still trying. But anyone whose products had to be put into large quantity production without any real pre-testing in the aircraft was heading for trouble right from the start, and he did not think that the blame for such trouble should be placed entirely on the oil cooler manufacturers.

Probably the most common cause of high oil temperatures in the Service was the phenomenon referred to as coring. There had been more muddled ideas and mistaken theories about that than about any other subject of which he could think. Mr. Nixon had thrown a good deal of light on it, but he had not emphasised strongly enough that coring resulted from the oil pressure drop being such that the relief valve opened and the cooler froze up. That was a phenomenon which could be found out on a prototype aircraft merely by fitting an oil pressure gauge at an appropriate spot on the cooler. But it was not done, and in one particular case it had taken a year to put the matter right. There was an old saying that "men reap as they sow," and he believed that that was partly the answer to the recent spate of oil cooler trouble.

Joining issue with Dr. Still's remarks concerning the secondary surface and honeycomb tube coolers, he said he did not know whether Dr. Still had intended to give the impression that the secondary surface oil cooler was the answer to the maiden's prayer when it came to coring or congealing; but the reverse was the case. If one considered the secondary surface cooler behind the water radiator, that was a horse of another colour; one should at least consider the honeycomb tube cooler in the same position. He emphasised that he was not defending the honeycomb tube cooler; he merely wanted to say that the disadvantage of the secondary surface cooler was the congealing problem experienced with designs so far tested.

MR. B. G. MARKHAM (Bristol Aeroplane Co., Ltd., Associate Fellow) congratulated Mr. Nixon on his excellent paper and on the work he had done for the Aircraft Industry, and acknowledged the help he had given to the Bristol Company.

Commenting that the heat-to-oil in the case of the Peregrine engine was about double as much, per B.H.P., as that of the Merlin, he said it would be interesting to know what developments had led to the lower figure for the Merlin engine.

On the general question of heat-per-engine B.H.P., it was very interesting to note the similarity of the percentages as between the different types of engine, the only outstanding exception being the Sabre, which paid a penalty for very compact design. However, perhaps the really interesting point was, not so much the heat-to-oil, but the size of oil cooler required for comparable installations; although for given heat-to-oil and given sizes of engines there was a very wide variation in the temperature rise of the oil circulated, the oil cooler sizes seemed to be about the same. He asked if Mr. Nixon could give figures.

Then Mr. Nixon had made rather a point of the variation of oil flows and had suggested that in some instances low oil flows had been adopted to avoid scavenging difficulties. That was one way of putting it.

Another way might be to say that engines with high rates of oil flow exhibited scavenging trouble, that there was trouble caused by large quantities of gas and oil coming out of the breathers when petrol dilution was used, and that larger circulation meant the introduction of larger and more complicated oil tank hot pots. However, if it were thought that there was advantage in having large circulations—and certainly there was for some coolers, if not for others—it was not difficult to provide. As a matter of fact, one of the reasons which Mr. Nixon had not mentioned for the wide variation in rates of circulation, was that on some engines the pressure pump relief oil, the oil not required by the engine, went into the sump and cooled down the other oil and increased the circulation.

In spite of Mr. Nixon's statement that it was difficult for some engine makers to increase the circulation, he could not see any great difficulty in adopting this system. The Bristol Company had done so experimentally, without having to scratch their heads very hard, and it had certainly worked quite well. If that were done, then by making the pump large enough they could have any rate of circulation they liked. They had discussed the question of circulation with some of the oil cooler manufacturers, and the number of answers received was almost as great as the number of people who were questioned. But it did appear that in some cases where a small cooler was adopted it had meant leaving out so many baffles that the cooler became weak ; and although dummy stiffeners could be put in, the general effect was to reduce the efficiency of the cooler. Large circulations did not help to reduce cooler size. However, the rate of circulation must be high enough to keep the outlet temperature to a reasonable figure.

If one were interested in achieving the lowest internal drag, probably a large cooler having a low rate of air flow through it would give the best result. But the drag was so small that it did not matter much. The great thing was to produce a compact unit which would fit into a wing or somewhere else where it would not be inconvenient.

Some experience had been gained in the use of high-temperature solders on coolers for Bristol engines; the results obtained with coolers made with such solders were superior to those obtained with others. There had been far less failures; and although at normal working temperatures high temperature solder might not be of great importance, under overload conditions it proved a considerable safeguard. The only failures experienced with coolers in which high-temperature solders were used were not dangerous failures; they were merely small leaks.

The omission of oil tubes from the American coolers in order to prevent coring in flight and to facilitate quick warming up, which was said to be done " by guess and by gum," seemed to result in making the air flow controls look silly and complicated. We should not be led away from the straight line for achieving air flow control; if we were led away we should find ourselves going round in a "viscous" circle. There was a great deal to be said for the control of air flow.

Noting with interest that Mr. Nixon had not mentioned vibration, he said a great many people had claimed that cooler failures were due to vibration. That seemed to be a reasonable suggestion. But personally he had never seen a failure of a cooler which could be definitely attributed to vibration. However, he would be interested to hear the views of others on the matter.

WING-COMMANDER S. B. BAILEY (R.A.F. Bomber Command, Associate Fellow), referring to the effect of power plant vibration on oil coolers, and Mr. Markham's view that not very many failures were caused thereby, recalled that early in the war considerable trouble was experienced in a certain aeroplane the radiators of which were more un-

reliable than any others in Bomber Command. It was noted that more trouble had occurred on the port outer power plant than on any other, and that on that particular power plant the vibration amplitudes were about three times as great as on the other installations. Grave complaints were made about the radiator trouble, and a new mounting was designed on scientific lines, using rubber in shear. As a result the radiator on that installation, instead of being the worst example in the Command, became the best. Therefore, he rather disagreed with Mr. Markham's view.

MR. MARKHAM pointed out that he was referring to the effect of vibration on oil coolers.

WING-COMMANDER BAILEY replied that he appreciated that. But a honeycomb oil cooler was very like a honeycomb radiator in construction, and he would say that vibration effects on a radiator might be taken as being similar to vibration effects on an oil cooler. Both had the same sort of soldered joints, and solder had notoriously weak fatigue properties.

With regard to the unreliability of oil coolers on air-cooled installations, he suggested that insufficient attention had been paid to the mounting of the coolers, for it would appear that the method of mounting had a big effect on cooler reliability. As showing how badly the air-cooled installations compared with the liquid-cooled in that respect, he said his experience was that on the air-cooled installations the oil cooler troubles were at least ten times as great as on the liquid-cooled. No doubt much of that difference was because of the higher oil outlet temperatures on the air-cooled engines; but he was quite sure that if the air-cooled power plant designer would incorporate a proper mounting for the oil cooler he could at least approach the degree of reliability attained by the oil cooler on the liquid-cooled engine.

The oil-cooler failures per 10,000 hours of flying on air-cooled engines were approximately equal in number to all other engine failures put together, and were far greater than the combined oil cooler and radiator failures of liquid-cooled engines. Inasmuch as oil cooler failure was an incipient cause of complete engine failure, the matter should be borne in mind.

MR. T. A. STERN ((Rolls-Royce, Ltd.), speaking as a Development Engineer concerned with oil coolers for liquid-cooled engines, discussed future developments. The immediate requirements, he said, were very well-known. We wanted lower weight, greater cooling efficiencies, lower cooling drag, decreased bulk, greater robustness, greater reliability, and so forth. Pleading for more development apart from the actual oil cooler itself, however, he said the present available lubricating oils had a very narrow temperature range; we needed oils having a very much wider temperature range over which they would retain the same specific properties as those within the present narrower range. If that position were achieved, so that engines could operate at very much lower oil temperatures and very much higher temperatures in the engines, many advantages would accrue. For example, at the cold end a wide temperature range had obvious advantages for cold starting.

Quoting some figures for heat-to-oil on a Merlin engine at different oil inlet temperatures, he said they applied to conditions of constant power output and constant cooling jacket temperature; the jacket coolant temperature was 120° C. The figures were as follows:—

Oil Inlet Temperature	H.P. Input to Oil
70° C.	56
90° C.	53
110° C.	42
130° C.	25
150° C.	Nil

The advantage of an oil having the properties of that at 150° C. was obvious. He saw no immediate prospect of the develop-

ment of such an oil, but there was a real need for it.

SQUADRON-LEADER C. H. POTTS (Associate Fellow), commenting that he had not appreciated that such a small advance had been made in connection with oil cooling during the past 15 years, said we seemed to be running round in a vicious circle. He felt that it had been a retrograde step when the honeycomb oil cooler was accepted for all types of engines, and more particularly the radial engine. An effort should have been made to find a more reliable type. There had, however, been a lack of co-operation between aircraft and engine manufacturers, both having followed the line of least resistance without due consideration of the vital part the oil cooler had to play.

On the one hand the aircraft designer accepted anything that could be stowed away conveniently, and on the other the engine designer had showed little or no interest, once having found a means of reducing the oil inlet temperature to an acceptable figure.

He asked if any information had been obtained regarding the effect on oil cooling of (1) putting the cooler in the suction circuit, and so preventing air passing through the cooler with the oil, and (2) putting the cooler in the scavenge outlet, but incorporating a device for removing the air from the oil (as far as reasonably possible), before it entered the cooler.

Finally, he thought that the oil pressure gauge was wrongly placed, with especial reference to the extremities of the engine, *i.e.*, the reduction gearing. Serious trouble had been experienced on radial engines in France in the cold winter of 1939/40, due largely to lack of oil in the reduction gear, with no means of knowing that the feed to the gears was blocked by congealed oil.

MR. P. G. LUCAS (Hawker Aircraft Limited, Fellow) believed that a number of oil cooler failures in flight could be attributed to high peak loadings of back-pressure. He had personally taken an active part in the flight development of a very large number of oil coolers and on numerous occasions had nearly had heart-failure when watching the cooler inlet back-pressure gauge. This frequently showed violent pressure fluctuations up to as high a figure as 100 lbs. when gliding with the throttle closed at high altitudes.

Because of this, his Company had been experimenting with automatic Air Flow controls with considerable success, as was shown by consistently steady and low back-pressures under all conditions of flight.

He could not say for certain that the use of automatic Air Flow controls had reduced the percentage of oil cooler failures as they had not as yet, sufficient collective experience, but at any rate, the results obtained were interesting and in his opinion, warranted more extensive use.

FLIGHT-LIEUT. M. C. HORNSBY said that if he were an aircraft constructor he would be very much concerned about the amount of heat that had to be dissipated in the oil cooler, and therefore, he would expect the engine constructors to give him very accurate figures. He would be very perturbed if he realised that as between different engines there was a possible variation of as much as 30 per cent. in respect of the heat picked up by the oil, and he would require a very accurate heat balance. He would attempt to do a heat balance in flight.

The fact that the degree of congealing varied so much as was indicated by Mr. Nixon would also worry him a great deal. He was not sure whether Mr. Nixon had meant that every oil cooler of a certain type varied, or whether each particular type of cooler had a different congealing characteristic. But he urged that a test should be inaugurated so that the congealing characteristic of a cooler could be ascertained within reasonable limits in the early stages of design.

MR. A. M. MARTIN (Serck Radiators, Ltd.) *Contributed*. The Americans have

standardised on circular and oval oil coolers. In this country the aircraft people have often been reluctant to accept circular coolers and have called for special shapes, which not only increase production difficulties, but also make the design problems more difficult.

He considered it most undesirable to control the oil outlet temperature by by-passing oil. There were two oil paths in parallel, and the arrangement tended to be unstable, since a small decrease in the oil flow through the cooler caused the oil in it to be cooled to a lower temperature. This in turn raised the pressure drop through the cooler and reduced still further the oil flow through it.

It was obvious that the American method of preventing coring by dummy tubes, internal leaks, etc., must lead to some loss of performance under warm air conditions. He would like to know what loss was normally tolerated for the sake of the improved cold air performance.

MR. K. C. HUNT (Intava, Ltd., Associate Fellow) *Contributed*. The author had throughout accepted the fact that the engine lubricating system and the oil cooler in particular must cater for oils of viscosity temperature characteristics substantially similar to those shown in Fig. 19 of his paper. This was, of course, the only attitude which he could take at the present time when on the one hand the engine builders insisted on the use of oils having a viscosity of around 20 centistokes at 210° F., and, on the other, the Petroleum Industry had so far been unable to produce lubricants from natural sources, showing a substantially smaller viscosity change with temperature than that represented in Fig. 19.

It would be interesting, however, to have the author's views as to the effect on the oil cooling system, and the attendant difficulties in maintaining satisfactory operation of it, of a reduction in viscosity from 16,000 centistokes to something between 1,000/2,000 centistokes at the temperature of 10° F. (−12½° C.) which was established in Fig. 19. If, as one had every reason to believe, the oil cooling problem would be materially eased both in terms of heat rejection and in mechanical reliability by such a reduction in viscosity, then there would seem to be every reason to try and achieve it.

There were two ways in which this could be done:—

One, by the use of oils of generally lower viscosity range, which might incidentally have a proportionately small viscosity temperature change; and the other by a material decrease in the viscosity temperature change (or increase in Viscosity Index) of oils of the 20 centistokes at 210° F. class.

Dealing with the former proposition in more detail:—

The engine builders had built up a technique of clearance design and surface finish and hardness as the result of many years of engine development on what were commonly called 100″ oils (*i.e.*, about 20 centistokes at 210° F.). There was, however, no reason whatever to suppose that the engine builder was incapable of developing his engine to use lower viscosity range oils of the order of 10 centistokes at 210° F., a typical example of which would have a viscosity at the temperature of 10° F. of 2,000 centistokes.

It was, however, obviously a matter for the engine builder to decide in consultation with the oil cooler manufacturer whether or not the benefits which could be gained in the cooler by operation on the lower viscosity oil would be justified by the amount of engine development work required.

The solution of the problem by the second method was in the hands of the Petroleum Industry and was of necessity a longer term project. In the writer's opinion, however, it was not unreasonable to beleve that within say, five years, 100″ oils might be available by synthesis, having a viscosity temperature characteristic such that at 10° F. their viscosity was between 1,000 and 2,000 cts.

Mr. Nixon could give a useful lead to

the Petroleum Industry by laying down the maximum oil viscosity at, for instance, 10° F. which would be really acceptable to the engine oil cooler.

MR. NIXON'S REPLY TO THE DISCUSSION.

Mr. Rowarth. First he expressed his appreciation of the remarks made by Mr. Rowarth, with whom he had greatly enjoyed working in his role of consultant to C.R.D. on heat exchanger equipment. There had been from time to time a keen struggle between the oil cooler manufacturing industry on the one hand and themselves on the other, but they had enjoyed it, and would continue to do so, and he hoped that in time the effort would bear fruit.

He was glad that Mr. Rowarth had emphasised his own often expressed opinion of the poor reliability of oil coolers. Although he had perhaps indulged in understatement in the written Paper, he had been more forthright in the spoken Lecture, and would like to repeat his words so that they might go on record " the behaviour of oil coolers in service is a subject which has given rise to more grief, alarm and despondency during the war than has been occasioned by any other component. Certain designs have, in fact, succeeded in reaching and staying at the epidemic level."

Commenting on Mr. Rowarth's deprecation of the use of soft solder, he considered that it was perfectly satisfactory material if it were used properly, and he saw no reason why it should be condemned in the majority of its present-day applications. He was aware that one or two air-cooled engine manufacturers were anxious to increase the oil outlet temperature to a figure which would be higher than the initial melting point of solder, but he had seen no evidence which condemned soft solder as a material. Most of the troubles were due to the improper application of the solder. In all the hundreds of failed oil coolers which he had had an opportunity of examining, there had not been a single instance of a soundly-made joint having failed.

The normal American oil cooler made with orthodox soft solder was completely reliable, and it was tested, without failing, at a temperature of 150° C.

The aluminium alloy oil cooler was not actually in universal employment in the United States, but there was no doubt that it had arrived, and would stay, supplanting the majority of brass/copper types.

Dr. Still. He wished to dispel any impression that the American designs were not entirely satisfactory from the heat transfer point of view. It was true that the methods of determining the optimum disposition of diffusion passages were empirical, and often necessitated a great deal of work in the cold wind-tunnel. But the cumulative effect of work of this kind, which had been exerted in the States, had resulted in a basic design of oil cooler which was far ahead of contemporary British designs, in respect of reliability and consistency of thermal behaviour. It must be remembered that British oil coolers as a whole were being considered, and primarily the honeycomb type, which had been in such universal use until recently. Compared with these, American honeycomb oil coolers had shown themselves to be far superior. At the same time they served to give point to his remarks concerning the potential reliability of soft solder and soft soldered construction, when properly applied.

He was glad that Dr. Still had made clear the advantages of the secondary surface type, and that he had further clarified the greater reliability of liquid-cooled engine systems relative to those of British air-cooled engines.

Mr. Lickley. He had referred to the wide variations in oil flow. He was afraid that there would always be wide differences between engines of diverse types and makes. The reasons had been dealt with in the

Paper, and were largely bound up with design traditions and idiosyncrasies. The figure quoted, of a $\pm 30\%$ variation in oil flow of a given type of engine had occasioned some comment, but in actual fact aircraft manufacturers and oil cooler designers had been coping with ranges of this and higher orders for a long time past. It just happened that this particular engine manufacturer, a leading American one, had been more honest and helpful than average. In fairness to his colleagues, however, he should state that the variation in the oil flow of the Merlin engine was a good deal smaller than this.

Regarding oil velocities and back pressures, this was a good point which opened up interesting possibilities. A simple treatment of the problem was not possible, as the effects of increased viscosity at low temperatures on pressures in the pipe lines, etc., had to be considered. Viewed solely from the cooling aspect, a high oil velocity in a cooler designed to withstand high pressures had advantages which were under active consideration and development at the present time.

The normal maximum back-pressure on existing coolers was of the order of 55 lb./sq.in., at which pressure the by-pass valve opened. The pressure drop through a Merlin oil cooler, at normal temperatures, was only about 12 lbs./sq.in.

There was very little surplus solder in the honeycomb oil cooler, but there could be a great deal in secondary surface types. One American aircraft manufacturer took the step of removing carefully all the surplus solder from a radiator and returning it to the supplier with his compliments.

Light alloy components had not been in service as long as soft-soldered types, but there was every indication that they would be at least as reliable as the best brass ones.

The omission of certain tubes from American oil coolers had only a slight effect on the maximum heat dissipation, but a beneficial effect on the overall heat dissipation under all conditions such as might otherwise produce local congealing. The heat dissipation characteristics of American oil coolers were closely specified by the Army-Navy authorities, and they were in fact little different from our own.

Mr. Markham. The main factor in reducing the heat-to-oil of the Merlin engine had been improved crankcase scavenging.

He felt that Mr. Markham was wrong in assuming that oil cooler sizes remained roughly the same regardless of the heat-to-oil. For a given tube diameter and length, the heat dissipation of an oil cooler varied as its frontal area, within fairly close limits. The frontal area of the oil cooler for the Merlin 24 Lancaster power plant was only .47 sq. ft. That for the Merlin 85 Lincoln power plant, with shorter tubes and designed for higher altitudes, was .72 sq. ft. The latest Hercules oil coolers had a frontal area of approximately 1 sq. ft.

Perhaps he had an unfair advantage in having had intimate experience of both air- and liquid-cooled engines. He saw no reason, however, to modify his statements regarding rate of oil flow of engines. There had definitely been a school of thought which favoured a low flow for an air-cooled engine and, in fact, examination of the figures on Table I showed a tendency for the specific oil flow to decrease on the later Bristol engines. The big disadvantage of a low flow was that in order to maintain the velocity of flow at an acceptable figure in the cooler to reduce risk of congealing, the designer was apt to become involved and to complicate the cooler by having so many passes of intricate shape that the distribution across the passes was upset, again leading to congealing troubles. This had been dealt with very fully by Mr. Wareing. (Ref. 21).

Regarding high temperature solders, since he had never yet seen a sound soft-soldered joint fail, he remained to be convinced of their value. We were dealing with components which had unfortunately varied in quality extremely widely, and great care should be exercised in interpreting the results

of comparative reliability tests on a small number of units.

Strong exception was taken to the reference to "small leaks." The only oil cooler which could be tolerated was one which had no leaks at all. In addition, evidence was accumulating that the majority of serious leaks began as small leaks.

He hoped he had not given the impression that he was against airflow control. Although some American oil coolers would decongeal without control of airflow, airflow control itself was quite a good thing.

Wing-Commander Bailey. In his experience he had seen very few cases where vibration had been the primary cause of trouble with oil coolers. The particular radiator referred to by Wing-Commander Bailey was of honeycomb type, it was much bigger than the average oil cooler, and it suffered severely from its mounting, which squeezed it. As it had not to be designed for the pressures obtaining in oil coolers, its casing stiffness was probably lower, and it suffered in consequence.

He welcomed Wing-Commander Bailey's frank and illuminating statement on the rates of failure in service. It was a matter which invariably roused his ire, and it was still proving difficult to make some people on the manufacturing side realise how serious it was.

Mr. Lucas. The remarks on back-pressure were interesting, as supporting the arguments in favour of decongealing oil coolers, complete by pass valves (*i.e.*, surge relief valves) and airflow control. He did not think that airflow controls were likely to reduce the number of failures appreciably at this stage, as so much of the trouble was due to poor manufacture. The wonder was that such poor joints as were so often revealed lasted as long as they did.

Mr. Steadman. He had some sympathy with Mr. Steadman's difficulties earlier in the war, when insufficient time was allowed for testing, but he felt that there could be no excuse for the continued failures with components of up-to-date manufacture. Recently 36 of the latest and best oil coolers of all makes and designs, for both liquid- and air-cooled engines, had been submitted to the latest M.A.P. test, which was more stringent than the test at present in use, and which would become mandatory at the beginning of 1946. Eight of these coolers had passed the test without showing any leaks. Those 8, together with 10 others which had developed leaks to only a small extent during this test, were then subjected to the American pressure-cycle test. After an average of 2,000 cycles on each cooler, the tests had to be discontinued, as the compressor could not keep pace with the leaks! The average number of pressure cycles to the first leak was 630, and the best result to first leak was 1980 cycles.

He had also subjected to this pressure cycle test two American coolers which were rejected by the USAAF because the valve gaskets were leaking, after operational service. The tests had been interrupted to enable him to bring the coolers along to the meeting as concrete evidence, after 30,000 cycles. These were second-hand coolers, of not very modern vintage, perhaps 12 months old. That experience should give the lie to those who claimed that it was not fair to subject soft solder to stresses at such temperatures (150° C.).

Mr. Stern. The figures given by Mr. Stern showing the reduction of heat-to-oil with increased oil outlet temperature, were extremely interesting as illustrating an unexpected advantage of the liquid-cooled engine. They were obtained by Wright Field on a Merlin engine, during tests with the oil cooler blanked off altogether, to determine the effect of cutting out the oil cooler in the event of flak damage.

Squadron-Leader Potts. The answer to the first point was that honeycomb oil coolers could be perfectly reliable, if properly made, instanced by the American examples, but that a secondary surface type had advan-

tages in this respect, at any rate when compared with contemporary British versions. The secondary surface oil cooler was only being used in series, as shown in the Merlin installation in Fig. 48, or in the Gallay mixed matrix. With present types of oil there would be serious congealing if it were used alone, on an air-cooled engine.

Regarding the position of the cooler in the oil system, he did not think it would be feasible to install it in the suction circuit in view of the pressure drops involved. The removal of air from the oil would be of all round benefit as reducing frothing, sludging and oxidation, but it might have a slight adverse effect on heat transference due to the reduction of turbulence in the oil.

Squadron-Leader Potts had raised a valuable point in connection with the quick establishment of flow to all parts of the engine. This was a point which would be seized upon by the protagonists of oil dilution.

Flight-Lieut. Hornsby. Replying to the pertinent comment on heat-to-oil and oil flow, he only wished that engine manufacturers were all as helpful as he hoped his own company was, and the particular American firm already quoted. He had had considerable difficulty in collecting figures which he felt could be relied upon, for inclusion in Table I.

Expressing agreement with Flight-Lieut. Hornsby concerning the congealing characteristics of cooler design, he hoped that the provision of an increasing number of cold wind tunnels would help considerably in enabling all the characteristics of oil coolers to be determined before they went into production. But, from all the failures he had seen, he still believed that every one had been due primarily to poorly soldered joints. There was no magic in the fact that American coolers could withstand 30,000 cycles in the pressure test, whereas ours could not withstand 700. It meant that throughout the 80 yards of soldered joint in the cooler all of it was sound. We had simply to raise the quality of the joints so that we could eliminate the few inches of weak joints which so often existed in British types. He was confident that the quality could be improved considerably, but we had a long way to go before we reached the American standard.

Mr. A. M. Martin. Several interesting points had been raised. The action of the Americans in standardizing circular and oval shapes had been of considerable advantage. The shapes permitted light rigid designs, and the stipulation of oil flow and heat dissipation characteristics in specification ANC 75a had allowed manufacturers to establish a range of sizes likely to satisfy any conditions that might be met. The problems of thermal design had undoubtedly been reduced, but in their place had been substituted a stringent thaw-out requirement entailing much wind-tunnel work.

From the point of view of mechanical reliability, the advantage of the circular shape was chiefly one of lightness. After all, any shape could be made rigid by adding a little extra weight in the form of stiffeners. It was in the detail design of such features as permit easy soldering, as shown in Fig. 44, that the Americans had scored, and in the exclusion of such handicaps as rivets. In connection with Fig. 44 an interesting point arose. This detail design had persisted in American oil coolers from the first, about 12 years ago, and 10 years ago it was being copied by Britain, but as native British designs were evolved the basic value of the feature was lost sight of, and it was dropped.

Regarding the control of oil temperature by by-passing, Mr. Martin had described a cycle leading to congealing and overheating by by-passing. It was unwise to generalise, however, as the situation could be modified by a sufficiently high relief valve setting, or by a thermostatic by-pass valve. Airflow control, under thermostatic control, was of course the final answer to Mr. Martin's point.

As already mentioned in reply to Mr.

Lickley, dummy tubes had only a small effect on performance. A quantitative answer was difficult as the factors involved were so numerous and different in the two countries, but an examination of the performance characteristics of British and American oil coolers was being made by M.A.P. in conjunction with the industry, which should provide an answer.

Mr. Hunt. A real service had been done in submitting his résumé of oil viscosity characteristics and their influence. In the author's opinion it was most encouraging to have Mr. Hunt's view that oils of lower viscosity were within sight, but it was difficult to lay down a minimum acceptable figure. Any reduction in viscosity at low temperatures, and particularly any reduction in the rate of increase of viscosity at low temperatures, would be an advantage in direct proportion to that reduction, in reducing the tendency to congeal and in permitting higher oil velocities without undue increase in pressure. The problems of water cooling were small compared with those of oil cooling. Purely from the cooling point of view, the closer we approached the viscosity characteristics of water, the easier would be our problem.

Oil was supplied primarily to meet the needs of the engine, however. The reduction in viscosity at high temperatures would result in an increased flow, but there would appear to be no great difficulty in coping with this. From all other aspects the reduction in viscosity and in viscosity index would appear to present an all-round gain. Cold starting would be greatly facilitated, reduction gears would receive their meed of oil in good time, de-aeration would be easier and frothing would be reduced. Presumably, should oil churning occur in a particular engine, less heat would be put into it by the work done on it. By the time such oils were available, however, there was every possibility that active development on piston engines would have ceased. The jet propulsion and i/c turbine units would welcome thinner oils to an even greater degree. The lubrication of these engines was required for the high speed bearings, and, in the case of the turbine driving an airscrew, by the reduction gearing. Here there would be an all-round gain, and it was hoped that the Petroleum Industry would need no further encouragement to embark upon the development of oils as described by Mr. Hunt.

THE CHAIRMAN proposed a vote of thanks to Mr. Nixon for his very comprehensive paper, which he believed would become a classic in relation to oil coolers. At the same time he expressed the hope that, as the result of the discussion of all the troubles that had been experienced with oil coolers in the past whether they were due to the engine manufacturer having had to put up with inferior workmanship from the cooler manufacturer, or whether the cooler manufacturer had not given the information as to what was really required and, therefore, could not be expected to supply what evidently the customer did not know he wanted—we should be able to sort out the problem !

BALL AND TAPER ROLLER BEARINGS

by

Dr. M. S FRENKEL

(Translated by P. Meyer, B.Sc.)

Dr. Frenkel is a theoretical physicist who has contributed original work to quantum physics. During the war he has applied his knowledge to the mathematical and physical treatment of problems confronting industry. He has, purely theoretically, without laboratory research, provided general solutions to engineering problems of which he had no previous experience, and these solutions have been experimentally confirmed on the designs executed by Dr. Frenkel.

This paper shows admirably the theoretical method of approach adopted by Dr. Frenkel in solving practical problems.

The author's note on the " Foundations of Theoretical Research " is given below :

This paper is intended to serve as a demonstration of the following general principles of pure theoretical research :

To solve theoretically a problem in, say, engineering means first of all to determine the relations of the occurrences for the type of machine in question, *i.e.*, to represent these occurrences in a unified picture.

In the light of this picture there emerge from the infinitely large number of possible permutations of constructional parts only those which produce the essential occurrences for a required result, in such a way that from the relations of the occurrences the relations of the dimensions of these essential constructional parts are obtained, while all other constructional parts are eliminated as superfluous and thus harmful.

In this way, just as, *e.g.*, mathematics shows the geometrical shape enclosing maximum volume with minimum surface-area to be a sphere, the construction for best results is obtained immediately with the minimum of constructional parts.

With these relations of the dimensions, developed on the basis of the unified picture of the occurrences, all those of the essential influences, which without the relations of the dimensions negatively affect the results, will also serve the attainment of the required result, *i.e.*, through the relations of the dimensions harmony will have been achieved.

The foundation of theoretical research must be looked for in the development of philosophy—in the pursuit of unification and harmony.

Only advantages which are derived in the advancement of unification and harmony are not accompanied or followed by disadvantages.

This kind of progress develops a content of one's own " processes of unification," which enlarges infinitely and never leaves one—which I would call the " gravitation of the spirit," because through this one continually develops the relations of further occurrences and in this way represents these occurrences in the " picture of unification," which enlarges infinitely—which leads to the advancement of unification and harmony

Synopsis

This analytical treatment gives general considerations concerning " true rolling " of a rolling body in a circular path : firstly, for constant speed of its centre of gravity, and, secondly, for acceleration of its centre of gravity. From these are derived the relations of the dimensions, as function of the conditions, and the corresponding constructions for both taper roller bearings and ball bearings, which are essential to produce " true rolling " at constant and variable shaft speeds.

In the case of taper roller bearings, it is shown that the centre of gravity of a roller must be in a specified position for true rolling to take place, this position being nearer the smaller end of the conical rolling surface than the larger end, in contrast to present constructions. Some constructions satisfying the above conditions are described.

For angular contact ball bearings, the relations of the dimensions derived include those of the track grooves.

Allowance is made for the fact that, in general, load distribution on the rolling bodies is very uneven, and constructions according to the relations of the dimensions are given which provide load distribution.

General Considerations

The method adopted in the analysis is first to consider an inelastic sphere under the minimum of influences giving motion in a circular path. Other influences are then added individually to the theory until the motion of the sphere reaches its motion in practice.

Case 1, Fig. 1, shows a sphere of radius r resting on a frictionless horizontal plane and held in the pocket of a frictionless guiding cage. The cage is fixed to a vertical shaft rotating with constant angular velocity ω_y and forces the sphere to describe a circular path only under the influence of centripetal force exerted by the frictionless guiding cage about the shaft (Y axis). The radius of the path described by the CG of the sphere is R and the velocity of the CG $V_c = R\omega_y$.

As there are no external forces acting on the sphere other than the centripetal force exerted by the cage which acts through the centre of the sphere, there will be no rotation of the sphere about any axis through its CG. Thus the velocities of all particles of the sphere relative to its CG will be zero, and the velocity vectors of all particles of mass will be equal and parallel to the velocity vector V_c of the CG. The sphere has pure motion of translation.

Under the centripetal force the velocity vector $V_c = R\omega_y$ rotates with angular velocity ω_y about the vertical Y axis, and thus also about the axis through the CG parallel to the Y axis. Since the relative velocities of all particles in the sphere with respect to the CG are zero, the velocity vectors of all particles must rotate with the velocity vector of the CG about the Y axis through the CG, though the sphere itself does not rotate about this axis.

Thus as the sphere moves, the velocity vectors of all particles rotate about an axis through the CG parallel to the Y axis with angular velocity ω_y relatively to the sphere.

Referring to Fig. 1, as the sphere does not rotate about the vertical axis through its CG, the line CA fixed in the sphere will move parallel to itself and in time dt will move from position C_1A_1 to position C_2A_2, the points C and A both having parallel and equal velocities $V_c = V_a = R\omega_y$ *[1], their paths being arcs of circles $C_1C_2 = A_1A_2 = R\omega_y dt$ with centres O and O_a respectively ; Thus, C_1A_1 and C_2A_2 are equal and parallel to the line $O\,O_a$. While CA is moving parallel to itself, the velocity vectors of C and A rotate relative to the sphere with angular velocity ω_y about the axis through the CG parallel to the Y axis.

At time t the point A_1 on the sphere

*[1] This case must be differentiated from the case where the sphere itself is fixed to a shaft rotating with angular velocity ω_y for, in this case, a particle at istance $(R+x)$ from the Y axis has velocity $(R+x)\omega_y$ and not $R\omega_y$.

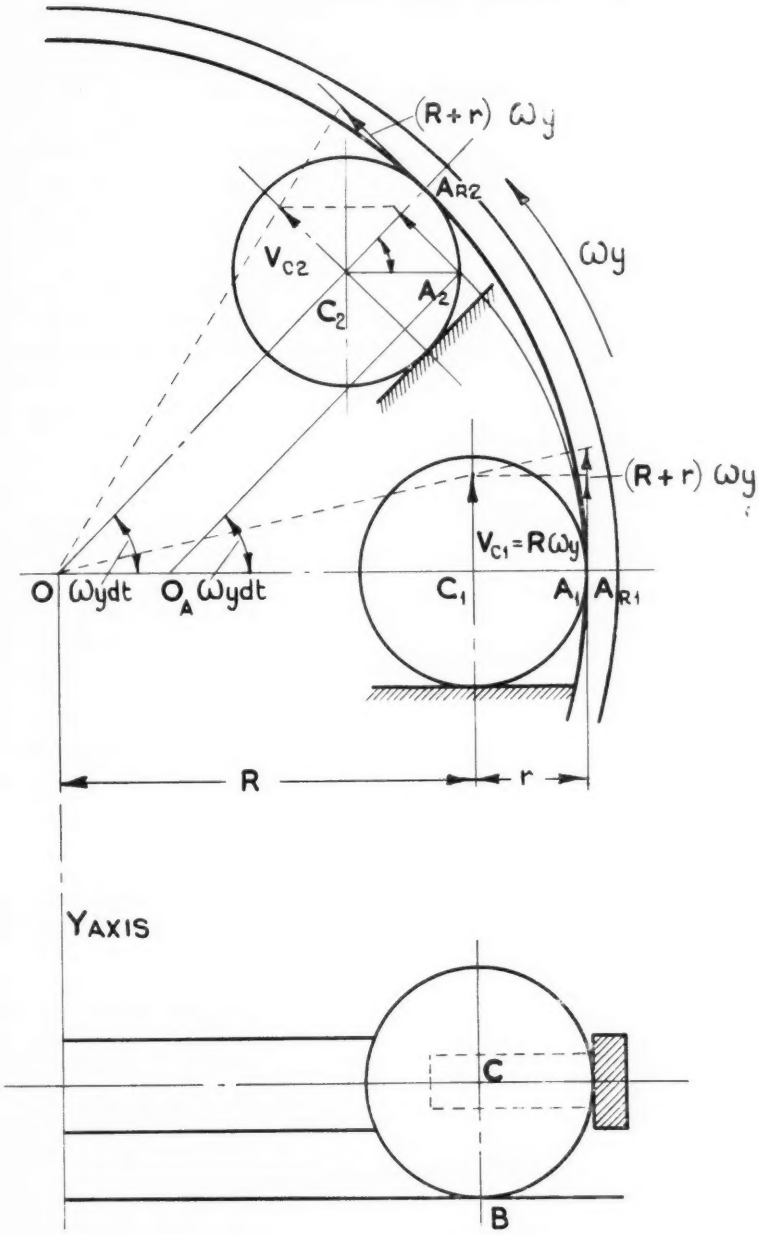

(R+r) ωy
AR2
ωy
VC2
C2
A2
(R+r) ωy
VC1=Rωy
O
ωydt
OA
ωydt
C1
A1
AR1
R
r
YAXIS
C
B

FIG. I

coincides with the point A_{r1} on the frictionless cage. At time $(t+dt)$ the point A_1 will have moved to point A_2, describing an arc with centre O_1 and radius R with velocity $R\omega_y$. The length of this arc A_1A_2 is $R\omega_y\,dt$.

In the same time the point A_{r1} describing its circular path with centre O and radius $(R+r)$ will have moved to A_{r2}. As its angular velocity is ω_y, the arc $A_{r1}\,A_{r2}=(R+r)\omega_y dt$ and the velocity of the point is

$$(R+r)\omega_y=R\omega_y+r\omega_y.$$

Thus the point A_{r1} on the cage has a velocity equal to the sum of the velocity of translation of the sphere $R\omega_y$ plus a velocity $r\omega_y$ due to a rotation with angular velocity ω_y about the vertical axis through the centre of the sphere. This point therefore rotates about this axis with angular velocity ω_y relative to the sphere. (A general proof of this is given in App. 1.)

The cage thus rotates with the same angular velocity ω_y about the vertical axis through the CG of the sphere as the velocity vector V_c of the CG.

Case 2.—If friction suddenly occurs between the external ring of the guiding cage and the sphere, the relative velocity of the cage and the sphere at their point of contact will produce the full sliding frictional force μC_f,—see Fig. 2—where C_f is the centrifugal force acting through the point of contact between sphere and cage, and μ is the coefficient of sliding friction. μC_f, together with the opposite force exerted on the sphere by the side wall of the guiding cage in front of the sphere[*2] forms the couple

$$\mu C_f r=I\,\frac{d\omega_y}{dt}$$

These conditions are shown in Fig. 2.

This couple will produce a rotation of the sphere about the vertical axis through its CG, but will vanish when the velocity of the point on the ball in contact with the cage is the same as the velocity of the point of contact on the cage and thus of value $(R+r)\omega_y$.

When this couple disappears the particles of the sphere will have an additional velocity value of ω_y due to rotation and thus due to friction on the external ring of the guiding cage, the sphere will have a motion of translation with velocity $R\omega_y$, and a rotation of angular velocity ω_y about the vertical axis through its CG which is equal to the angular velocity of the CG of the sphere about the Y axis.

Thus in this case the velocity vectors of all particles of the sphere rotate with the same angular velocity about the vertical axis through the CG as the sphere itself, and so the velocity vectors may be considered to be fixed in the sphere.

Case 3.—The third case to be considered is when there is no friction between the outer guiding ring and the sphere but where there is friction between the sphere and the horizontal plane. The sphere has a pure motion of translation in a circular path and there is a force T acting parallel to the Y axis towards the plane through the CG of the sphere, as shown in Fig. 2*a*.

If friction suddenly arises between the sphere and the horizontal plane, there will be a relative velocity V_b at the point of contact B between the plane and the sphere, all points of which are initially moving with velocity V_c. At the instant of friction $V_b=V_c$. This will produce the full sliding friction force $\mu_s T$ (where μ_s is the coefficient of sliding friction) acting in the opposite direction to V_c. This friction force with the opposite force[*3] through the CG exerted on the sphere by the side wall of the guiding cage behind the sphere, forms a couple $\mu_s T.r$

[*2] μC_f causes a pressure between the sphere and the side wall of the cage in front of the sphere—not shown in Fig. 2—producing a reaction from this wall and thus couple $\mu C_f.r$.

As the cage is fixed to the shaft rotating with constant angular velocity ω_y, V_c remains constant—there is no dynamic reaction.

[*3] $\mu_s T$ causes a pressure between the sphere and the side wall of the cage behind the sphere, producing a reaction from this wall and thus couple $\mu_s T.r$.

As the cage is fixed to the shaft rotating with constant angular velocity ω_y, V_c remains constant—there is no dynamic reaction.

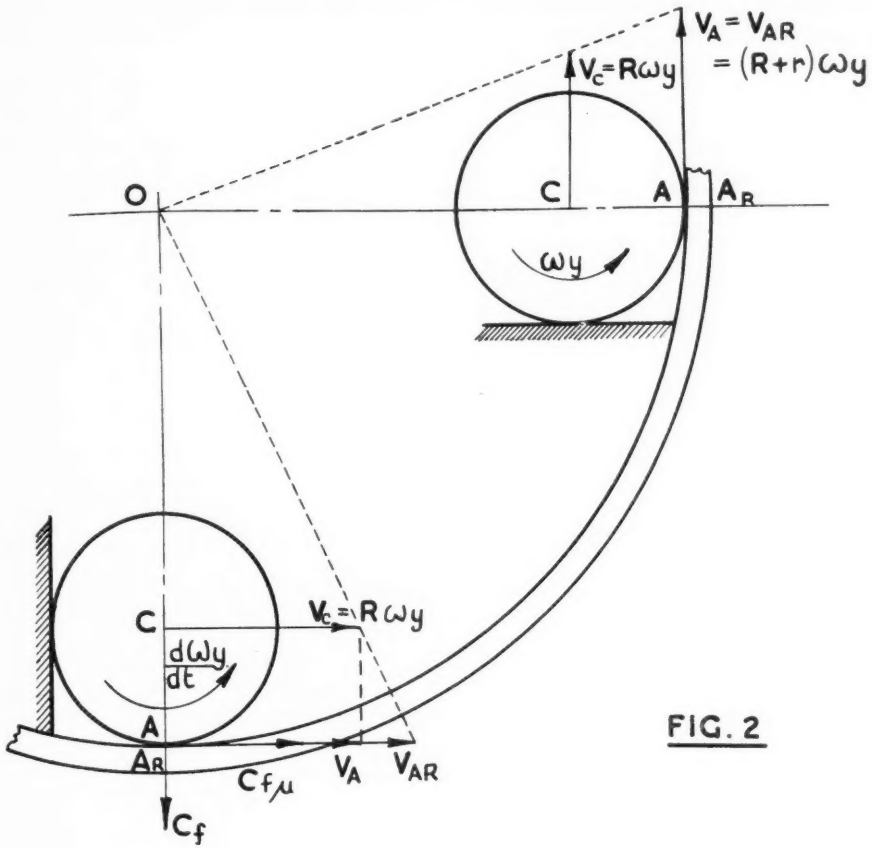
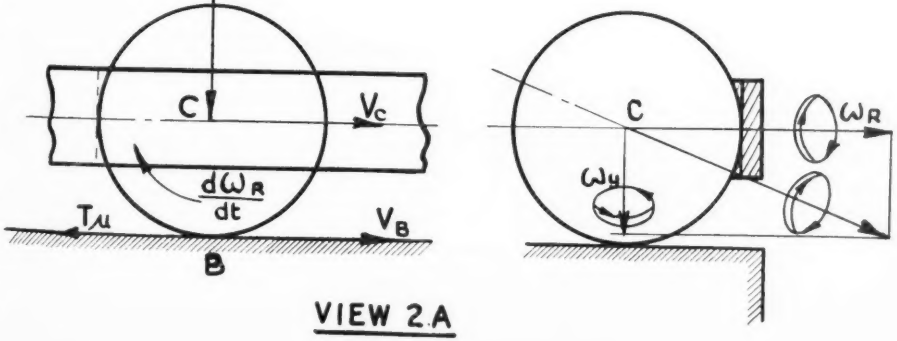

$V_A = V_{AR}$
$= (R+r)\omega y$
$V_c = R\omega y$
O
C
A
A_R
ωy
C
$V_c = R\omega y$
$\frac{d\omega y}{dt}$
A
A_R
$C_f\mu$
V_A
V_{AR}
C_f
T

FIG. 2


C
V_c
$\frac{d\omega_R}{dt}$
$T\mu$
V_B
B
C
ω_R
ωy

VIEW 2A

in the plane containing velocity vector V_c and the point of contact B. This couple will produce an angular acceleration about the axis through the CG normal to the plane containing V_c and B (about the rolling axis), given by

$$\mu_s T.r = I\frac{d\omega_r}{dt}$$

Now the plane of the couple $\mu_s T.r$, as it contains vector V_c, rotates with angular velocity ω_y about the axis parallel to the Y axis through the CG (the axis BC), while the sphere itself does not rotate about this axis, as there is no couple acting in the plane normal to BC to produce such a rotation. Thus the couple $\mu_s Tr$ will never act in the same plane of the sphere but will at every instant produce angular acceleration in a plane rotating through the sphere so that no angular velocity ω_r of the sphere will arise. This is because for this angular velocity to be produced it would be necessary for the couple to act for a time t in the same plane of the sphere so that

$$I\omega_r = \int_0^t \mu_s T.r.dt.$$

Therefore in this case there will always be a relative velocity at the point of contact between the sphere and the plane, and the full sliding friction force will act there.

Case 4.—The final case to be considered in this section is when friction arises between the sphere and both the plane and the guiding ring.

Initially the sphere has pure motion of translation in a circular path with angular velocity ω_y and the cage has angular velocity ω_y about the Y axis. As has been shown the point on the cage in contact with the sphere has velocity $(R+r)\omega_y$ while the instant friction arises, the point of contact on the sphere will have velocity $R\omega_y$ due to its motion of translation. As shown in Case 2 the couple $\mu C_f.r$ will form, causing angular acceleration about the Y axis through the CG (axis BC), and this couple will continue to act until the relative velocity between the sphere and the cage will have vanished, i.e., when the sphere will have the same angular velocity ω_y about axis BC as the point of contact on the cage has about this axis. This is the same angular velocity as the CG has about the Y axis, and thus, as already shown in Case 1, the same angular velocity ω_y as velocity vector V_c has about axis BC, so that the sphere will get the same angular velocity ω_y about BC as the velocity vector V_c has about this axis. Thus velocity vector V_c will remain in the same plane of the sphere.

The couple $\mu_s T.r = \dfrac{I\,d\omega_r}{dt}$ caused by the relative velocity at the point of contact B between the sphere and the plane will be in the plane of V_c and V_b and will thus always lie in the same plane of the sphere (because V_c is in the same plane of the sphere).

In this way the couple $\mu_s T.r$ will continue to act in the same plane of the sphere until the angular velocity ω_r about the rolling axis (normal to the plane of V_c and V_b) has reached such a value that the relative velocity between sphere and plane at their point of contact B is zero,

$$V_b = V_c - r\omega_r = O,$$

and thus B will become the instantaneous centre of motion.

Thus, for true rolling in a circular path, a sphere must have two angular velocities about two axes at right angles*[4] where each angular velocity is produced by a couple in the plane normal to the axis of this angular velocity.

The following requirements for true rolling can thus be established :—

1. In order that the velocity vector of the CG, V_c should always be in the same plane

*[4] The general considerations concerning the Gyroscopic Inertia Couples which arise with these two angular velocities about two axes at right angles will be given in a following paper, together with their applications to Ball Bearings, Ball and Conical Cup Type Governors, etc.

of the sphere normal to the rolling axis, so that the rolling axis should be fixed in the sphere, the angular velocity of the sphere about the axis (BC) through its $C.G.$ and its point of contact with the rolling surface (which is to become the instantaneous centre of motion), must be equal to the angular velocity of the velocity vector V_c of the C.G. about this axis.

If this is not the case, V_c will not remain in the same plane of the sphere, the rolling axis will move relative to the sphere, and true rolling will be impossible.

2. The sphere must have such an angular velocity about the axis through its $C.G.$ normal to the plane containing the velocity vector V_c and the point of contact (B) with the rolling surface (about the rolling axis), that the relative velocity (V_b) between sphere and rolling surface at the point of contact (B) should be zero,

$$V_b = V_c - r.\omega_r = O$$

i.e., the point of contact (B) should be the instantaneous centre of motion.

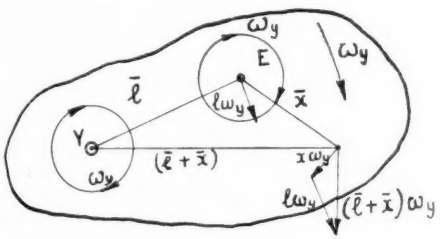


Fig. 3

It follows from this that the two angular velocities of the sphere about the axes at right angles to each other will result in a general angular velocity of the sphere about an axis through the CG, making a certain angle with the rolling axis, at which there will be true rolling.

Appendix 1. General Proof

If a body rotates about an axis Y with angular velocity ω_y, then any point E in the body at normal distance l from the Y axis will have velocity $l\omega_y$, and any other point at a distance $\bar{x}$ from the axis parallel to the Y axis through E will have velocity

$$(l+\bar{x})\omega_y = l\omega_y + \bar{x}\omega_y.$$

Thus it can be stated that every particle in the body has a velocity equal and parallel to the velocity of point E ($l\omega_y$), plus a velocity ($\bar{x}\omega_y$), due to a rotation with angular velocity ω_y about an axis through E parallel to the Y axis. Thus if a body rotates about an axis Y fixed to it, all particles of the body rotate with the same angular velocity ω_y about a Y axis through any other point fixed relatively to the body which gives each particle its velocity due to this rotation and apart from that, all particles have velocity parallel and equal to the velocity of this point (*e.g.*, the velocity $l\omega_y$ of point E).

About any other axis through E inclined at angle α_2 to the Y axis, all particles rotate with angular velocity $\omega_y \cos \alpha_2$.

TAPER ROLLER BEARINGS

General Remarks

The method is first to determine the angle between the axis of the general angular velocity of the roller and its geometric axis as function of the dimensions for the condition that " true rolling "*5 of the roller on the fixed race exists.

The next stage is to give general considerations concerning true rolling during acceleration of the centre of gravity of the rolling body.

The third stage is the determination of the relations between the forces on the roller which arise during changes of the shaft speed due to the rotating and fixed races respectively, as function of the dimensions, for the condition that true rolling of the roller on the fixed track exists during changes of the shaft speed.

Then the relations of the dimensions are determined such that both the above condi-

*5 " True rolling " denotes rolling on a generating line of the roller in the contact area between fixed track and roller.

tions should be obtained, *i.e.*, such that true rolling of the roller should exist at constant and variable shaft speed.

It is shown that the relations of the dimensions essentially require a certain position of the centre of gravity relative to the conical rolling surface of the roller for true rolling to take place.

This position in general is such that the distance of the centre of gravity from the inner (smaller) end cross section of the conical rolling surface of the roller is less than its distance from the outer end cross section—which is in contrast to present constructions.

It is shown that for a position of the centre of gravity which is not the one required by the relations of the dimensions, no true rolling can exist—skewing of the rollers due to the couples acting on them arises.

Of many possible constructions which give a required position of the centre of gravity, one is suggested which calls for the drilling of a conical hole through the roller in the form of a convergent divergent nozzle, because this has the added advantage of making the bearing act as a centrifugal pump for air, and thus assist in its own cooling.

The extension of the relations of the dimensions to provide also for distribution on the rollers of a radial load on a bearing will be given in a later paper.

Theory

Section I. — Consider the arrangement shown in Fig. 4. If the axis of the general angular velocity (ω) of the roller through its centre of gravity makes an angle $\hat{\phi}$ with the geometrical axis of the roller, the value of angle $\hat{\phi}$ must be determined at which there will be true rolling of the roller on the fixed race.

The component of the general angular velocity of the roller in the direction of the geometric axis will be $\omega \cos\phi$, and $\omega \sin\phi$ will be the angular velocity of the roller about the axis normal to the geometric axis which passes through the centre of gravity and intersects the generating line of the roller in contact with the fixed race track at A_2.

If the angular velocity of the centre of gravity of the roller about the bearing (Y) axis is denoted by ω_c, then ω_c is also the angular velocity of the velocity vector V_c of the centre of gravity about the Y axis through the centre of gravity. Hence $\omega_c \sin\alpha_2$ is the angular velocity of the velocity vector V_c of the centre of gravity about the axis A_2C, where α_2 is the angle between the geometric axis of the roller and the bearing (Y) axis.

For true rolling, as has been shown, the velocity vector of the centre of gravity (V_c) must always lie in the cross section of the roller, *i.e.*, be normal to the geometric axis of the roller. This means that the angular velocity $\omega \sin\phi$ of the roller about the axis A_2C must be the same as the angular velocity $\omega_c \sin\alpha_2$ of the velocity vector (V_c) of the centre of gravity about this axis, so that

$$\omega \sin\phi = \omega_c \sin\alpha_2 \quad . \quad . \quad (1)$$

For, if

$$\omega \sin\phi \lesssim \omega_c \sin\alpha_2,$$

the velocity vector V_c will not remain in the cross section of the roller normal to the geometric axis, skewing of the roller with oscillations will arise :

In the cross section of the roller there will only be a component of the velocity vector V_c of the centre of gravity, which will oscillate in magnitude,*[6] and thus no true rolling can then be possible.

Thus, in order that the velocity vector V_c should always lie in the cross section of the roller,

$$\omega \sin\phi = \omega_c \sin\alpha_2 \quad . \quad . \quad (1)$$

from which

$$\omega = \omega_c \frac{\sin\alpha_2}{\sin\phi}.$$

As the component of the general angular

*[6] The component of velocity V_c in the direction of the geometric axis of the roller will cause oscillating dynamic pressures against the flange.

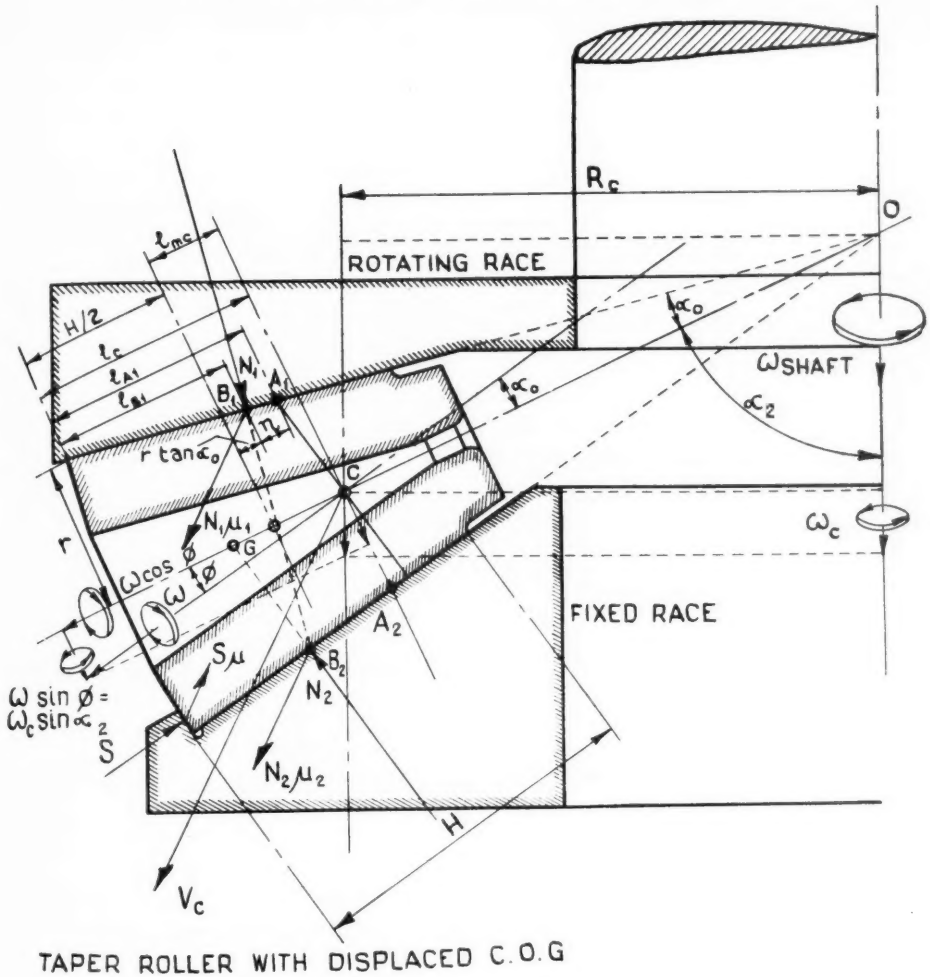


FIG. 4

velocity of the roller in the direction of the geometric axis of the roller is $\omega\cos\phi$, and for true rolling on the fixed race the relative velocity V_{a2} at the point of contact A_2 between the roller and the fixed race must be zero, *i.e.*, A_2 must be the instantaneous centre of motion, and the generating line of the roller in contact with the fixed race must be the instantaneous axis of motion of the roller, therefore

$$V_{a2}=V_c-r_c\,\omega\cos\phi=0 \quad . \quad (2)$$

where r_c is the radius of the roller in the cross section of its centre of gravity.

Now,

$$V_c=R_c\omega_c,$$

where R_c is the radius of the circle described by the centre of gravity of the roller about the bearing (Y) axis, and, as seen from the geometry of Fig. 4,

$$R_c = r_c \frac{\sin \alpha_2}{\tan \alpha_o},$$

where α_o is half the apex angle of the conical roller, and α_2 is the angle between the geometric axis of the roller and the bearing (Y) axis. Thus

$$V_c = R_c \omega_c = r_c \frac{\sin \alpha_2}{\tan \alpha_o} \omega_c \quad . \qquad (3)$$

Substituting in (2) for V_c from (3) and for ω from (1):

$$V_{a2} = r_c \frac{\sin \alpha_2}{\tan \alpha_o} \omega_c - r_c \omega_c \frac{\sin \alpha_2}{\tan \phi} = O,$$

from which

$$\tan \phi = \tan \alpha_o \quad . \quad . \quad . \qquad (4)$$

Thus the essential condition for true rolling of the roller is that the axis of the general angular velocity of the roller should make an angle ϕ with the rolling axis, which is equal to α_o, half the apex angle of the conical roller, and further that the general angular velocity vector should have such a direction that its component along A_2C, normal to the rolling axis, should be equal in direction as well as in magnitude to the component along A_2C of the angular velocity ω_c of the centre of gravity of the roller about the bearing axis.

Fig. 4 shows this condition fulfilled for a bearing in which the fixed race carrying the flange is the outer surface of a cone, while Fig. 4 (*a*) shows a bearing in which the fixed race carrying the flange is the inner surface of a cone.

In this second case it is seen that the component of the general angular velocity vector of the roller in the direction of the

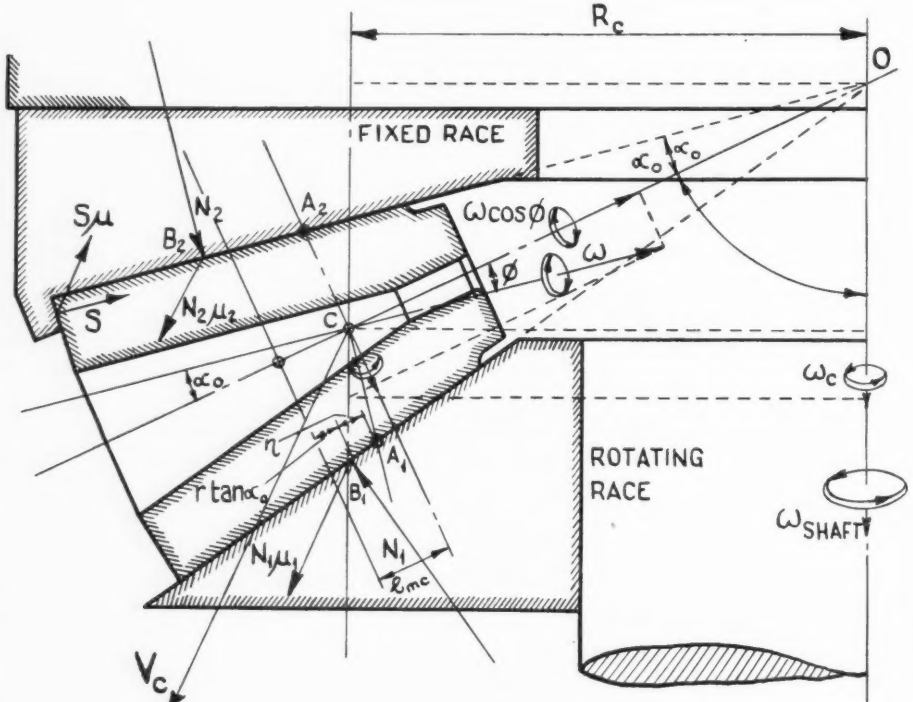


FIG. 4 A

rolling axis is reversed, but that the component of the general angular velocity vector of the roller in the direction of A_2C is the same in direction and magnitude as the component of the angular velocity ω_c of the centre of gravity of the roller about the bearing (Y) axis, so that the condition for true rolling is also given. The direction of the general angular velocity vector is seen to be $(180° - \alpha_o)$ to the geometric axis.

Section II.—The next stage is to develop the relations of the dimensions so that the angle ϕ should also in fact become equal to angle α_o, *i.e.*, relations of the dimensions must be developed such that the vector of the angular acceleration of the roller arising during changes of the shaft speed,*[7] which develops the general angular velocity, should make this angle α_o with the rolling axis.

Now, the angular acceleration of the roller (treated in further detail later) is mainly produced by the couple formed during changes of the shaft speed by the friction force $N_1\mu_1$ parallel to V_c, due to the resultant N_1 of the pressure forces between the rotating race and the roller, together with other forces parallel to V_c, and the dynamic reaction at the centre of gravity, which arise due to $N_1\mu_1$.

These couples must produce an angular acceleration whose vector should coincide with the general angular velocity vector making an angle α_o with the geometric axis, as required for true rolling.

This is only possible when the component of this angular acceleration in the direction A_2C has the same sense of rotation as the component about the axis A_2C of the angular acceleration of the centre of gravity $\frac{d\omega_c}{dt}$ about the bearing (Y) axis.

For this to be possible, the centre of gravity must be in such a position relative to the conical rolling surface of the roller that because of the friction forces parallel to V_c, acting on the roller during changes of the shaft speed from the rotating and fixed races respectively, there should be exerted in the plane normal to the axis A_2C, a couple with this sense of rotation.*[8]

From this requirement, as will be treated at length later, the position of the centre of gravity of the roller must be such that the plane (A_1C) through the centre of gravity C normal to the required general angular velocity vector, should intersect the generating line of the rotating race at a point A_1, such that the point of action B_1 of the resultant N_1 of the pressure forces between the rotating race and the roller should lie between point A_1 and the external (remote from the shaft) end cross section of the conical rolling surface of the roller. This is seen with reference to Fig. 4.*[9]

Thus the normal distance l_{a1} of point A_1 from this external end cross section of the conical rolling surface should be

$$l_{a1} > l_{b1},$$

where l_{b1} is the normal distance of the point of action B_1 of the resultant N_1 from the same external end cross section of the rolling surface.

In the further development following, the distance l_{a1} will be exactly determined

$$l_{a1} = f(r, H, \alpha_o, \ldots).$$

From the condition $\tan\phi = \tan\alpha_o$, the distance l_c of the centre of gravity in its new position from the external end cross section of the conical rolling surface is given by

$$l_c = l_{a1} + (r - l_{a1} \tan\alpha_o) \tan\alpha_o . \qquad (5)$$

where r is the radius of the external end cross section of the conical rolling surface. The centre of gravity will be displaced to a posi-

*[7] There is actually only a constant mean angular velocity of a shaft, but no constant angular velocity of a shaft due to fly wheel effects, etc.

*[8] In the case of orthodox taper roller bearings, where the centre of gravity lies between the central and the external end cross sections of the conical rolling surface, the sense of rotation of the couples in the plane normal to A_2C due to $N_1\mu_1$ and the other forces parallel to V_c will be opposite to the sense of rotation of the component along A_2C of the angular acceleration $d\omega_c\,dt$ of the roller centre of gravity about the shaft axis, which produces skewing.

*[9] This will emerge clearly from the later exact determination of the required position of the centre of gravity.

tion such that, during changes of the shaft speed, the above conditions will be fulfilled, while during constant shaft speed the couples due to the forces parallel to V_c will balance.

It will be shown exactly later in the paper that the centre of gravity will have a certain position as function of the dimensions,

This required displacement of the centre of gravity can be achieved by means of various constructions.

Before l_c is determined as function of the dimensions, that relation between the forces P_1 and P_2 parallel to V_c, which during changes of the shaft speed act on the roller

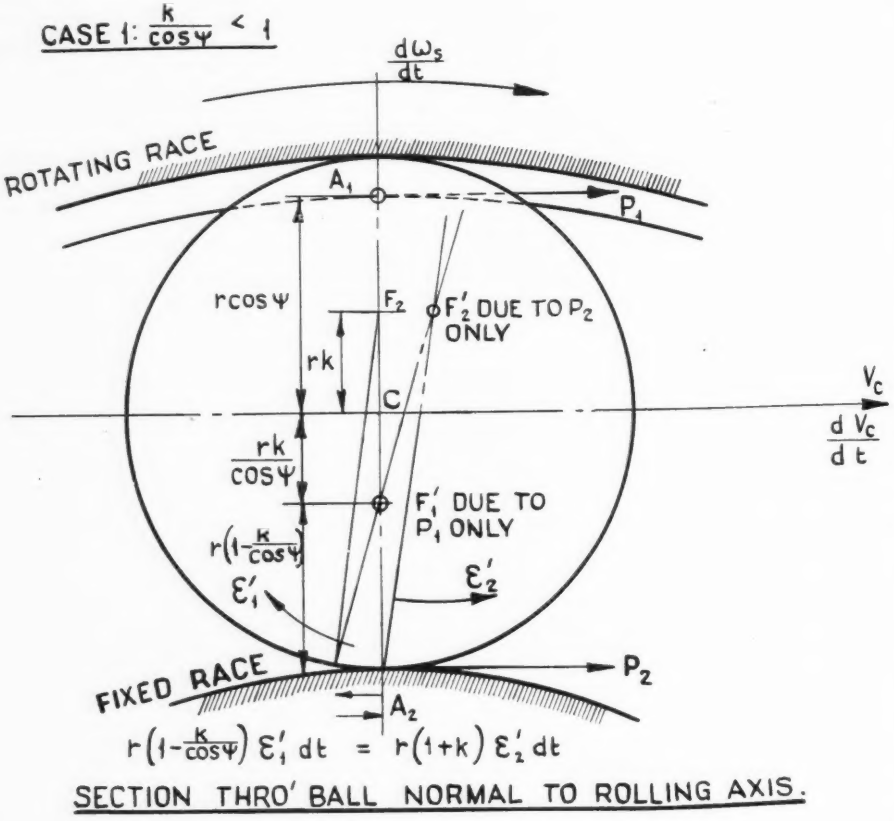


FIG. 5.A

$l_c=F(r, H, \alpha_o, \ldots.)$, which is located between the central cross section of the conical rolling surface and the inner (nearer to shaft) end cross section of this rolling surface. This is in direct contrast to the present forms of construction adopted.

from the rotating and fixed races respectively, must be determined, at which true rolling will also exist during changes of the shaft speed. (See Fig. 5*a*.)

P_1 is here defined as the friction force which the rotating race exerts on the roller

parallel to V_c during changes of the shaft speed, exclusive of the friction force required to overcome " rolling resistance."

P_2 is defined as the resultant friction force which the fixed race exerts on the roller parallel to V_c during changes of the shaft speed, including the friction force exerted by the flange, but excluding the friction force required to overcome " rolling resistance."

General Considerations

True Rolling during Acceleration of the Centre of Gravity of a Rolling Body.—Consider first the effect of the driving force P_1 only, where P_1 is in the direction of V_c for V_c increasing, and acts on the rolling body at point A_1 a distance $d=r\cos\varphi$ above the centre of gravity, as indicated in Fig. 5*a* (*i.e.*, as if there were no friction at the rolling surface).

The driving force P_1 together with its dynamic reaction at the centre of gravity—given by $P_1=Ma_1$, where a_1 is the acceleration of the centre of gravity due to P_1 only—will form the couple on the rolling body

$$P_1 r\cos\varphi=Ma_1 r\cos\varphi=I\varepsilon_1 \quad . \qquad (6)$$

where r is the radius of the rolling body at the cross section of the centre of gravity,

M the mass of the rolling body,

I its moment of inertia about the geometrical axis,

ε_1 the angular acceleration due to P_1 only.

Putting $I=Mr^2k$,

$$P_1 r\cos\varphi=Ma_1 r\cos\varphi=I\varepsilon_1=Mr^2k\varepsilon_1,$$

and hence

$$a_1=r.\varepsilon_1 k/\cos\varphi \quad . \qquad (6a)$$

This means that the point which has no acceleration at all due to P_1 and thus is the point through which the axis of the angular acceleration ε_1 passes—the instantaneous centre of change of motion F_1—is situated on the line CA_2 a distance $r\,k/\cos\varphi$ below the centre of gravity, as shown on Fig. 5*a*.

Thus, while the acceleration of the centre of gravity due to P_1 is

$$a_1=\varepsilon_1 r\,k/\cos\varphi$$

and the acceleration of the point of action A_1 of the driving force P_1 will be $r(1+k/\cos\varphi).\varepsilon_1$ in the direction of increasing V_c, the point A_2 in contact with the rolling surface will have the acceleration $r(1-k/\cos\varphi)\varepsilon_1$ in the opposite direction to V_c due to force P_1 only. If there were no friction at the rolling surface, the velocity of point A_2 relative to the rolling surface—which for true rolling should be zero—would be given an increase of $r(1-k/\cos\varphi)\varepsilon_1.dt$ opposite to V_c.

The question is, how large the friction force P_2 exerted by the rolling surface on the roller in the direction of V_c must be, in order that it should produce such an angular acceleration ε_2 in the opposite direction to ε_1, and thus an acceleration of the point A_2 in the direction of V_c, resulting in an increase in the velocity of point A_2 in the direction of V_c equal to $r(1-k/\cos\varphi)\varepsilon_1\,dt$, that the velocity of point A_2 relative to the rolling surface should remain zero.

The friction force P_2 together with its dynamic reaction at the centre of gravity. $P_2=Ma_2$, will form a couple

$$P_2r=Ma_2r=I\varepsilon_2=Mr^2k\varepsilon_2 \quad . \qquad (7)$$

Here the acceleration of the centre of gravity due to P_2 is $a_2=rk\varepsilon_2$, and therefore, for the angular acceleration ε_2 due to P_2 only, the axis, and thus the instantaneous centre of change of motion F_2, will lie on CA_1 a distance of $r.k$ above the centre of gravity.

The point of contact A_2 between roller and rolling surface, being at distance $r(1+k)$ from F_2, will have the acceleration $r(1+k)\varepsilon_2$ in the direction of V_c, which will produce the increase in the velocity of point A_2 of $r(1+k)\varepsilon_2\,dt$ in the direction of V_c.

Now, in order that the resultant increase in the velocity of point A_2 due to forces P_1 and P_2 should be zero,

$$r(1-k/\cos\varphi)\varepsilon_1\,dt=r(1+k)\varepsilon_2\,dt,$$

from which

$$\frac{\varepsilon_2}{\varepsilon_1}=\frac{(1-k/\cos\varphi)}{1+k} \quad . \qquad (8)$$

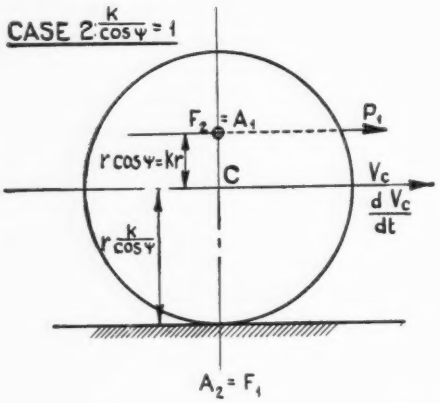


FIG. 5B

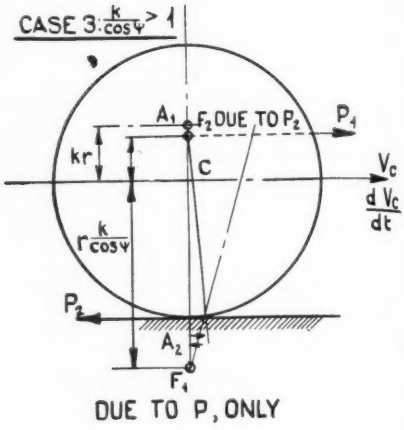


FIG. 5C

From equations (6) and (7),

$$\frac{P_2}{P_1}=\frac{(1-k/\cos\varphi)\cos\varphi}{1+k} \quad . \quad (9)$$

For a deceleration of the shaft, the same result is obtained in a similar way, only P_1 and P_2 as well as the acceleration of the centre of gravity will be in the opposite direction to V_c.

From this formula, we obtain three different cases :—

$$(1)\ \frac{k}{\cos\varphi}<1\ ;\ (2)\frac{k}{\cos\varphi}=1\ ;\ (3)\ \frac{k}{\cos\varphi}>1\ ;$$

(1) $k/\cos\varphi<1$ is the case where the line of action of the driving force P_1 parallel to vector V_c is above the instantaneous centre of change of motion F_2 due to friction force P_2, $r\cos\varphi>r\,k$.

This is shown in Fig. 5*a*.

Here the point of contact A_2 with the fixed race lies a distance $r(1-k/\cos\varphi)$ below the instantaneous centre of change of motion F_1 due to P_1, and due to P_1 will tend to have an acceleration $r(1-k/\cos\varphi)\varepsilon_1$ in the direction opposite to the acceleration of the centre of gravity. Therefore the friction force P_2, which for true rolling prevents a relative velocity due to this acceleration arising, will act in the same direction as the acceleration of the centre of gravity, and thus as force P_1, giving

$$P_2=P_1\frac{(1-k/\cos\varphi)}{1+k}\cos\varphi.$$

(2) $k/\cos\varphi=1$ is the case where the line of action of the driving force P_1 parallel to V_c passes through point F_2, a distance $r.k$ above the centre of gravity, $r\cos\varphi=r.k$.

This is shown in Fig. 5(*b*).

Then the instantaneous centre of change of motion F_1 due to P_1 will coincide with the point of contact A_2, the instantaneous centre of motion for true rolling. Thus for true rolling no frictional force at the rolling surface will be required.

$$P_2=P_1\frac{(1-k/\cos\varphi)}{1+k}\cos\varphi=O.$$

Thus, for example, for a ball, for which $k=0.4$, this condition can be attained for angle $\varphi=66°\ 30'$. For other types of rolling bodies, k can be made larger by suitable alterations in shape to attain this condition.

(3) $k/\cos\varphi.>1$ is the case where the line of action of the driving force P_1 parallel to V_c passes below the instantaneous centre of change of motion F_2 due to P_2 at the rolling surface, $r\cos\varphi<rk$. This is shown in Fig. 5(*c*).

Then, from the general expression,

$$P_2=P_1\frac{(1-k/\cos\varphi)\cos\varphi}{1+k}$$

$$=-P_1\frac{(k/\cos\varphi.-1)\cos\varphi,}{1+k}$$

i.e., P_2 is in the opposite direction to P_1 and thus to the acceleration of the centre of gravity.

This is because here the instantaneous centre of change of motion F_1 due to P_1 is situated below the point of contact A_2 by a distance $r(k/\cos\varphi-1)$, and therefore the point A_2 on the rolling body tends to have an acceleration $r(k/\cos-1)\varepsilon_1$ in the direction of the acceleration of the centre of gravity. Thus the friction force P_2 for true rolling must be in the direction opposite to the acceleration of the centre of gravity and thus opposite to the direction of P_1.

Section III.—Now, returning to the treatment of taper roller bearings:

For a taper roller with the usual small apex angle, the forces P_1 and P_2 are taken to act at the mean radius r_m, so that Case I of the above three cases applies. Because $\cos\varphi=1$, the formula reduces to

$$P_2=P_1\frac{1-k}{1+k} \quad . \quad . \quad (9a)$$

For this relation of the forces P_1 and P_2 arising during changes of the shaft speed, the instantaneous centre of motion will remain at A_2 even during changes of the shaft speed, the forces P_1 and P_2 both being in the direction of the respective acceleration of the centre of gravity.*[10]

Now, the force P_1 in the direction parallel to V_c, exerted by the rotating race on the roller during changes of the shaft speed (exclusive of the force required to overcome "rolling resistance"), and the force P_2 parallel to V_c exerted by the fixed race during changes of the shaft speed, including the friction force exerted by the flange on the fixed race, making contact with the roller end near its periphery as shown in Fig. 4, but excluding the force required to overcome "rolling resistance," form couples with their dynamic reactions at the centre of gravity. These produce the angular acceleration λ_1 about the geometric axis of the roller given by

$$(P_1-P_2)r_m=I_1\lambda_1. \quad . \quad (10)$$

where I_1 is the moment of inertia of the roller about its geometric axis, and r_m is the mean radius of the roller—while about the axis A_2C through the centre of gravity normal to the geometric axis of the roller these couples produce the angular acceleration λ_2 given by

$$(P_1+P_2)(\eta+r_{a1}\tan\alpha_0)+L=I_2\lambda_2. \quad (11)$$

In this equation I_2 is the moment of inertia of the roller about the axis A_2C through the centre of gravity normal to the geometric axis, and

$$P_1(\eta+r_{a1}\tan\alpha_0)$$

is the couple formed by P_1 and its dynamic reaction at the centre of gravity (in the plane normal to axis A_2C). Here

$$\eta=l_{a1}-l_{b1}$$

is the projection on to the geometric axis of the line connecting points B_1 and A_1.

B_1 is the point of action of the resultant N_1 of the pressure forces between the rotating race track and the roller. A_1 is the point at which the plane through the centre of gravity normal to the axis of the general angular velocity ω required for true rolling, intersects the generating line of the roller in contact with the rotating race track. l_{b1} and l_{a1} are the normal distances of these two points B_1 and A_1 respectively from the larger (external) end cross section of the conical rolling surface, and r_{a1} is the radius

*[10] As $P_1\leqslant N_1\mu_s$, where μ_s is the coefficient of sliding friction, and P_2 is smaller still, this force P_2 will, as will be proved, be exerted with the aid of the frictional force between fixed race track and roller, if the relations of the dimensions and construction for true rolling as developed in the following are adhered to, and under the condition that, as will be derived later,

$$\left(\frac{1-k}{1+k}+2\tan\alpha_o+\frac{C_f\sin\alpha_i}{N}\right)\leqslant 1,$$

where C_f is the centrifugal force.

of the cross section of the roller containing point A_1, where

$$r_{a1}=r-l_{a1}\tan\alpha_0,$$

r being the radius of the larger (external) end cross section of the roller.

In equation (11) the couples due to the forces acting from the fixed race—which forces form P_2, but which do not act at one point—and their dynamic reactions at the centre of gravity, are represented by two couples, namely,

$$P_2(\eta+r_{a1}\tan\alpha_0)+L,$$

where L also takes into account the couple in the plane normal to A_2C due to the forces required to overcome "rolling resistance." This couple L will be treated more fully later.

Thus, from the two equations (10) and (11)

$$\frac{I_2\lambda_2}{I_1\lambda_1}=\frac{P_1(1+P_1/P_2)[\eta+r_{a1}\tan\alpha_0]+L}{P_1(1-P_1/P_2)r_m} \qquad (12)$$

Now, in order that there should be true rolling, the following conditions must be fulfilled :—

(1) The axis of the angular acceleration *11 of the roller arising due to changes of the shaft speed must coincide with the axis of the general angular velocity ω of the roller which, as required for true rolling, makes an angle $\phi=\alpha_0$ with the geometric axis of the roller, and therefore

$$\left.\begin{aligned}\lambda_1&=\lambda\cos\alpha_0\\ \lambda_2&=\lambda\sin\alpha_0\end{aligned}\right\} \quad . \quad . \quad (13)$$

(2) The second condition for true rolling during changes of the shaft speed is, as already shown,

$$\frac{P_2}{P_1}=\frac{1-k}{1+k} \quad . \quad . \quad (9a)$$

so that the factors in the above equation (12) become

$$1+P_1/P_2=\frac{2}{1+k},\ \text{and}\ (1-P_1/P_2)=\frac{2k}{1+k} \quad (14)$$

Substituting from all the above conditions in equation (12)

$$I_2=\frac{I_1}{\tan\alpha_0\cdot r_m k}\left[r_{a1}\tan\alpha_0+\eta_1+\frac{L(1+k)}{2P_1}\right] \qquad (15)$$

The moment L, which is a function of the position of the flange, etc., will be represented precisely in the following developments.

As regards the required displacement of the centre of gravity of the roller, equation (5) gives the relation for the new position of the centre of gravity

$$l_c=l_{a1}+r_{a1}\tan\alpha_0 \quad . \quad . \quad (5)$$

where $l_{a1}=l_{b1}+\eta_1$

and $r_{a1}=(r-l_{a1}\tan\alpha_0)=r-(l_{b1}+\eta_1)\tan\alpha_0$

Distance l_{b1} giving the position of the resultant of the pressure forces between the rotating race and the roller has yet to be determined.

Section IV.—In connection with this determination of l_{b1} and the determination of the moment L, the positions of the points of action of the resultants N_1 and N_2 of the pressure forces between rotating race track and roller, and between fixed race track (excluding flange) and roller respectively during changes of the shaft speed, have to be determined. These positions will not

*11 The angular acceleration referred to is not the "general" angular acceleration of the roller, which consists of two components :

One, referred to above, arises during changes of the shaft speed and for a bearing constructed according to the relations of the dimensions serves only to alter the magnitude of the general angular velocity of the rolling body and will coincide with the general angular velocity vector as required for true rolling.

The second component, however, only serves to alter the direction of the general angular velocity of the rolling body, but not its magnitude, and is therefore normal to the plane containing the general angular velocity vector and the bearing (Y) axis. This means that in this case this second component of angular acceleration produces the rotation of the general angular velocity vector about the bearing axis with the same angular velocity $\omega_c+(d\omega_c\,dt)$ as that of the rotation of the centre of gravity of the rolling body about the bearing axis.

This second component of angular acceleration, which only changes the direction of the general angular velocity vector must therefore be normal to the general angular velocity vector, and will always exist because the general angular velocity vector always changes its direction (rotates).

This second component of angular acceleration causes the gyroscopic inertia couples which, as shown exactly in a later paper, are nothing but the couples of the centrifugal forces on the particles of the rolling body. In that paper gyroscopic effects in application to ball bearings and ball and conical cup type governors will be treated.

be the same as the positions of the points of action of resultants N_1 and N_2 at constant shaft speed.

Because of the static equilibrium of the roller in the plane containing the geometric axis of the roller and the bearing axis, the forces N_1 from the rotating race track, N_2 from the fixed race track and the force exerted by the flange must from their different points of action always intersect in one point.

Taking moments about the point of intersection G of the resultant N_2 with the geometric axis,

$$(N_1+N_2)(\tan\alpha_0-\mu_3)r\cos\alpha_0=N_1\cos\alpha_0 . n-Q \quad . \quad (16)$$

where the L.H.S. of this equation is the moment about point G of the reaction S from the flange—where

$$S=(N_1+N_2)\ (\tan\alpha_0-\mu_3)$$

for the cases as indicated in Figs. 4 and

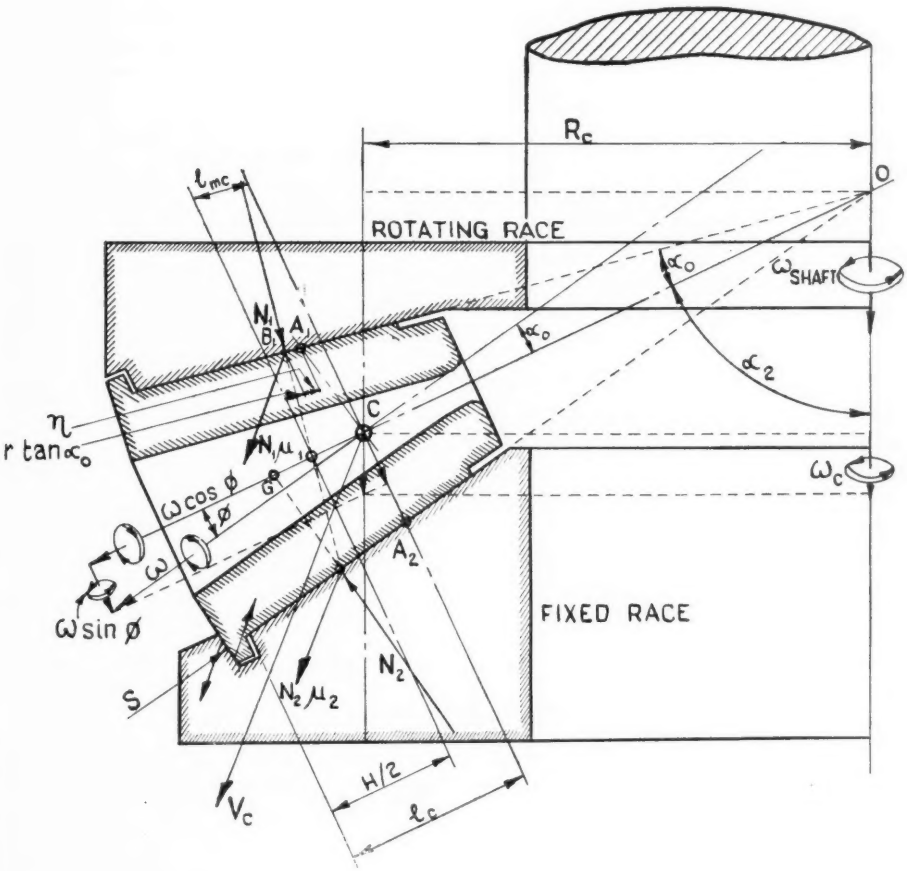


FIG. 6

6. The force from the flange on the fixed race in Fig. 4 acts near the periphery of the roller end (near the instantaneous axis of motion), and in Fig. 6 the construction shown is such that the force from the flange acts along the generating line of the fixed race. Further, n is the distance between the projections of the points of action of N_1 and N_2 on to the geometric axis, and Q is the gyroscopic inertia couple, whose influence has been neglected at the moment.

Now, in the direction of A_2C, normal to the geometric axis,

$$(N_1-N_2)\cos\alpha_0=(N_1+N_2)(\tan\alpha_0-\mu_3)\sin\alpha_0 \qquad (17)$$

From (16) and (17) by eliminating N_1 and N_2,

$$n=\frac{2r\ (\tan\alpha_0-\mu_3)}{1+(\tan\alpha_0-\mu_3)\tan\alpha_0} \quad . \qquad (18)$$

For $\alpha_0 \leqq 5°$, $[1+(\tan\alpha_0-\mu_3)\tan\alpha_0] \sqrt{1}$, so that

$$n=2r\ (\tan\alpha_0-\mu_3) \quad . \qquad (18a)$$

Substituting (18a) in (16).

$$N_1=N_2$$

μ_3 is the coefficient of friction between the race tracks and the roller in the direction of the generating lines of the roller, and its values during changes of the shaft speed differ from its value during constant shaft speed.*12

At constant shaft speed its value will nearly be the coefficient of "sticking friction," because in the direction of V_c, at right angles to the generating lines of the roller, there are practically no friction forces—only those required to overcome "rolling resistance." This, however, is only the case for true rolling.

Thus the angle α_0 can be chosen such that

$$\tan\alpha_0 \leqslant \mu_3.$$

In that case, from (18a)

$$n=0.$$

In this case, then, the resultants N_1 and N_2 will act at the centre of gravity of the generating lines of the surfaces which come into contact with the roller (across the track; for interrupted contact across the track, etc.), and in the case of uninterrupted contact across the track, N_1 and N_2 will act at the points bisecting the generating lines of the surface which comes into contact with the roller, apart from small displacements due to the gyroscopic inertia couple.

However, during changes of the shaft speed, when between both race tracks and the roller surface there will arise frictional forces parallel to V_c (*i.e.*, at right angles to the generating lines), the coefficient of friction along the generating lines will become negligibly*13 small, and then, as seen from (18a), n will not be zero.

Then, as seen with reference to Fig. 4, the resultant N_1 will move from the centre of the generating line of the contact surface towards the shaft, and the resultant N_2 will move from that centre in the direction away from the shaft.

As, however, the resultants N_1 and N_2 are equal in magnitude for the angle $\alpha_0 \leqq 5°$ N_1 and N_2 will be displaced symmetrically about the centre of the generating line by a distance of $n/2$.

From equation (18a). Then

$$n/2=r\tan\alpha_0 \quad . \quad . \qquad (18b)$$

Thus, the displacement of the resultant N_1, of the pressure forces between the rotating race and the roller (and thus also of the force P_1 parallel to V_c) during changes of the shaft speed from the centre of the generating line towards the bearing axis is

$$n/2=r\tan\alpha_0,$$

where r is the radius of the external end cross section of the conical rolling surface.

Thus, the distance of the point of action of the resultant N_1 between the rotating race and the roller during change of the

*12 In general, in the steady state, only the mean angular velocity for a revolution is constant, but the actual angular velocity of the shaft varies cyclically—it oscillates about the mean value.

*13 This will emerge clearly later from the determination of the values of μ_3 for various conditions.

shaft speed, from the larger (external) end cross section of the rolling surface is

$$l_{b1}=l_s+r \tan \alpha_0,$$

where l_s is the distance of the centre of gravity of the generating line of the surface of contact between races and roller from the larger end cross section.

In the special case of uninterrupted contact across the track

$$l_{b1}=H/2+r \tan \alpha_0,$$

where H is the height of the conical rolling surface of the roller.

From this

$$l_c = H/2+r \tan \alpha_0+r_i+r_{a1} \tan \alpha_0,$$

where $r_{a1}=(r-l_{a1}.\tan \alpha_0.)$

Thus the distance l_{mc} of the centre of gravity of the roller from the cross section containing the centre of gravity of the generating line of the rolling surface, will be

$$l_{mc}=\tan \alpha_0(r+r_{a1})+r_i \quad . \quad . \quad (19)$$

Section V.—Now, before continuing with the determination of l_{mc}, the limiting conditions within which the fixed race track can supply such a friction force that during changes of the shaft speed true rolling should be obtained, are derived. To obtain these limiting conditions, the case requiring maximum friction between fixed race track and roller is analysed, *i.e.*, the case of a maximum rate of increase of the shaft speed. This is done for the case of a bearing as shown in Fig. 4, with the flange acting on the roller end near its periphery.

On the one hand, for true rolling during changes of the shaft speed,

$$P_2=P_1 \frac{1-k}{1+k} = N\mu_s \frac{1-k}{1+k} \quad . \quad (9a)$$

where $N\mu_s$ is the frictional force between the rotating race track and the roller, it being assumed that in the limiting case of maximum acceleration of the shaft the value of the coefficient of friction approaches μ_s, the coefficient of sliding friction.

On the other hand, P_2, as the resultant of the forces parallel to V_c acting from the fixed race on the roller—the flange is taken as in Fig. 4 to make contact with the roller end near the periphery—must act in the direction of P_1 for true rolling, and for the case of an increase of shaft speed is given by

$$P_2=N\mu_2-(2N \tan \alpha_0+C_f.\sin \alpha_2)\mu_s \quad (20)$$

where $N\mu_2$ is the friction force of the resultant of the pressure forces between the fixed race track and the roller, with coefficient of friction μ_2, which is to be determined. In this equation further

$$(2N \tan \alpha_0+C_f \sin \alpha_2)\mu_s$$

is the frictional force exerted by the flange on the roller, and this for an increase of shaft speed acts in the opposite direction to P_2 and thus also to P_1.

C_f is the centrifugal force.

Now, from equations (9) and (20),

$$N\mu_2=P_2+(2N\tan \alpha_0+C_f\sin \alpha_2)\mu_s$$
$$=N\mu_s\left(\frac{1-k}{1+k}+2\tan \alpha_0+\frac{C_f\sin \alpha_2}{N}\right)$$

Now the force P_2 can only arise when $\mu_2 \leqslant \mu_s$, the coefficient of sliding friction—which from the above is only possible when

$$\frac{1-k}{1+k}+2\tan \alpha_0+\frac{C_f\sin \alpha_2}{N} \leqslant 1 \quad . \quad (21)$$

This is the extreme case, for which the above condition (21) gives the maximum value of $\mu_2 < \mu_s$—for all other cases (decrease of shaft speed, etc.) the required value of μ_2 will be smaller.

Section VI.—Now, in connection with the fact that during changes of the shaft speed the resultants N_1 and N_2 of the pressure forces between the rotating race track and roller and between the fixed race track and roller, respectively, are displaced by $n/2=r \tan \alpha_0$, as already shown, it is now possible to determine exactly the moments L in the plane normal to the axis A_2C.

For the case of a bearing, as shown in Fig. 4, where the flange makes contact with the sphered roller end near its periphery, *i.e.*, the resultant of the pressure forces between the flange and roller end is made to act near the instantaneous axis of motion:

$$I_2=\frac{Mr_m}{\tan\alpha_0}\left[l_{mc}+\frac{1+k}{2}\left(2\tan\alpha_0+\frac{C_f\sin\alpha_2}{N}-\frac{2k}{1+k}\right)r_m\tan\alpha_0-\frac{(1+k)H}{4}\left(2\tan\alpha_0+\frac{C_f\sin\alpha_2}{N}\right)\right] \quad (22)$$

This formula gives the value of the moment of inertia I_2 about the axis A_2C through the centre of gravity for substitution of the moment L for shaft speed increasing, and thus secures true rolling of the roller during acceleration of the shaft.

In order that, however, true rolling should also take place during a deceleration of the shaft, the expression for the moment of inertia I_2 on substitution of the moments for deceleration of the shaft, should have to give the same value for I_2 as it does for the case of the acceleration of the shaft.

Now, I_2 is a function of l_{mc}, the distance of the centre of gravity of the roller from the central cross section of the conical rolling surface, given by

$$l_{mc}=\tan\alpha_0(r+r_{a1})+\eta \quad . \quad (19)$$

where only $\eta \geqslant 0$ can be varied.

Thus, in order that the expression for the moment of inertia I_2 should give the same value of I_2 for a deceleration of the shaft as for an acceleration of the shaft, it is necessary, as can be proved, that

$$\eta_i=\frac{1+k}{2(1-k)}\left(2\tan\alpha_0+\frac{C_f\sin\alpha_2}{N}\right)H-\tan\alpha_0(2r_m+r_{a1})\geqq 0. \quad . \quad . \quad (23)$$

Thus, altogether, there are for the relations of the dimensions of the type of taper roller bearing, as shown in Fig. 4, where the flange on the fixed race makes contact with the sphered roller end near its periphery, these equations :—

$$I_2=\frac{Mr_m}{\tan\alpha_0}\left[l_{mc}+\frac{1+k}{2}r_m\tan\alpha_0\left(2\tan a_0+\frac{C_f\sin\alpha_2}{N}-\frac{2k}{1+k}\right)\right]$$
$$-\frac{Mr_m(1+k)H}{\tan\alpha_0.4}\left(2\tan\alpha_0+\frac{C_f\sin\alpha_2}{N}\right) \quad . \quad . \quad (22)$$

$$\eta_i=\frac{1+k}{2(1-k)}\left(2\tan\alpha_0+\frac{C_f\sin\alpha_2}{N}\right)H-\tan\alpha_0(2r_m+r_{a1})\geqq 0. \quad . \quad . \quad (23)$$

$$l_{mc}=\frac{1+k}{2(1-k)}\left(2\tan\alpha_0+\frac{C_f\sin\alpha_2}{N}\right)H-r_m\tan\alpha_0 \quad . \quad . \quad . \quad (19)$$

$$\left[\left(\frac{1-k}{1+k}\right)+2\tan\alpha_0+\frac{C_f\sin\alpha_0}{N}\right]\leqq 1 \quad . \quad . \quad . \quad . \quad (21)$$

$$r_m=r-\frac{H}{2}\tan\alpha_0$$

$$r_{a1}=r-\left(l_{mc}+\frac{H}{2}\tan\alpha_0.\right)$$

These equations, together with a further one concerning $\hat{\alpha}_0$, which will be developed now, give the relations of the dimensions for the above type of taper roller bearing, at which true rolling is assured during changes of the shaft speed as well as at constant shaft speed.

From these equations, the position of the centre of gravity relative to the central cross section of the conical rolling surface of a roller, together with its moments of inertia, are determined such that the axis of the angular acceleration of the roller arising due to changes of the shaft speed coincides with the axis of the general angular velocity of the roller as required for true rolling, so that true rolling will be obtained for acceleration and deceleration of the shaft as well as for constant shaft speed.

There are many methods of achieving constructionally the required position of the centre of gravity. One method is to make

the roller with a hole symmetrical about the geometric axis, if required in conjunction with other methods—such as adding a projection towards the shaft beyond the inner end cross section of the conical rolling surface—where this hole through the roller should have the shape of a convergent divergent nozzle, as indicated in Figs. 4 and 6. Thus, with the bearing in operation, such holes in the rollers simultaneously serve to move air for the cooling of the bearing, and even allow the bearing to be used as a blower.

In order that, for true rolling at constant shaft speed, the "rolling resistance" should be as small as possible, it is important that the roller should roll on the outside of a cone surface formed by the fixed race track, while it is driven by the inner surface of the cone formed by the rotating race track, as shown in all figures.

Section VIII.—Now, during changes of the shaft speed there are, apart from the rolling resistance, the above mentioned friction forces parallel to V_c between the race tracks and the roller (acting in directions normal to the generating lines of the race tracks), so that the frictional coefficients in the directions along these generating lines will be negligibly small.*14

Hence in this case the pressure exerted by the roller end on the flange will be

$$2N(\tan\alpha_0-\mu_3)+C_f\sin\alpha_2$$
$$\rightarrow 2N\tan\alpha_0+C_f\sin\alpha_2, \text{ where } \mu_3\rightarrow 0.$$

Thus there will be a pressure force between the flange and the roller end. However, the couple in the plane normal to axis A_2C due to the friction of this pressure force, will not produce skewing but, as shown in the previous developments, will, for the moment of inertia I_2 as given by the above relations, serve to give the roller the required angular acceleration about the axis A_2C. (This will be equal to the angular acceleration of the velocity vector V_c about the axis A_2C, for true rolling.)

As regards a state of constant shaft speed,*15 the following has to be stated with regard to constructions of taper roller bearing not made in accordance with the relations of the dimensions derived, and constructions which are in accordance with these relations.

For constructions of taper roller bearings as produced hitherto, the required relations of the dimensions are not observed:—

Firstly, the centre of gravity of the conical roller is situated between the larger (away from the shaft) end cross section of the rolling surface and the central cross section, instead of between this central cross section of the rolling surface and its smaller (near to shaft) end cross section, etc., etc.

Hence there is not only no true rolling, and thus always sliding friction between the roller and the race tracks in the direction of the velocity vector V_c of the centre of gravity, so that the flange always has to take up the pressure force

*14 The following development will illuminate this:

The sliding friction force on a body acts in the direction of the resultant relative velocity at the point of contact between this body and the surface upon which it slides. Thus the components of the sliding friction force in two directions normal to each other are proportional to the components of the resultant relative velocity in these directions.

How large, now, must an initial force P normal to the resultant relative velocity V be, in order to produce in the direction normal to V a displacement, and thus an acceleration a and an infinitely small velocity $a.dt$?

The new resultant relative velocity $(V+a.dt)$ will be in a different direction to V, and from the above consideration the friction force in the direction of force P will have the proportion to the friction force in the direction of V of $a.dt$ $V=0$. Thus the force P at right angles to the original relative velocity V will have to overcome no friction force as long as it has not produced a finite velocity in the direction normal to the original relative velocity V of sliding.

In contrast to this, if the centre of gravity of the body moves with velocity V, but there is no relative velocity at the point of contact between it and the plane, *i.e.*, the body rolls truly on the plane, then an initial force P normal to the velocity V of the centre of gravity will have to overcome the total sticking friction force in order to produce a small displacement, because then the resultant relative velocity at the point of contact between body and plane would be in the direction of this displacement.

*15 "Steady state" denotes a state in which the mean value of the shaft speed is constant—actually the angular velocity of the shaft in a steady state always varies about this mean value in a cycle between maximum and minimum values.

$$2N \tan \alpha_0 + C_f \sin \alpha_2,$$

but, what is more important, because the roller has neither the same angular velocity about axis A_2C as the velocity vector V_c of its centre of gravity, nor, during changes of the shaft speed, the same angular acceleration as the velocity vector V_c of the centre of gravity about axis A_2C, therefore this velocity vector V_c of the centre of gravity will not lie in the cross section of the roller—so that in addition to the above mentioned static pressures large dynamic pressures on the flange will be produced, which will oscillate (impact).

Through this increase of the pressure against the flange by the dynamic pressure, which due to impact is much larger than the static pressure, the friction forces between the flange and the roller end will be much increased, causing further loss of energy and skewing, on top of the energy losses due to sliding friction in the load carrying contact areas.

In constrast to this, for constructions according to the relations of the dimensions derived above, as soon as a constant shaft speed is attained*[16] there will only be rolling resistance to be overcome, requiring only very small friction forces from the races parallel to V_c, *i.e.*, normal to the generating lines of the race tracks. Therefore along the generating lines of the race tracks the frictional coefficient μ_3 will nearly be the coefficient of "sticking friction" (nearly 0.1), so that the pressure force against the flange will then be

$$S = 2N(\tan \alpha_0 - \mu_3) + C_f \sin \alpha_2, \quad \text{where } \mu_3 \geqslant 0.1 \quad . \quad . \quad (24)$$

If the half apex angle α_0 of the roller cone is chosen so that

$$2N \tan \alpha_0 + C_f \sin \alpha_2 \leqq 2N\mu_3, \quad \text{where } \mu_3 \geqslant 0.1 \quad . \quad . \quad (24a)$$

the pressure forces against the flange, which during changes of the shaft speed will be

$$S = 2N \tan \alpha_0 + C_f \sin \alpha_2$$

will become zero at constant shaft speed, $S = 0$.

Then the points of action B_1 and B_2 of the resultants N_1 and N_2 of the pressure forces between the race tracks and the roller will move to the centres of the contact surfaces (generally speaking, to the centres of gravity of the generating lines of the surfaces in contact)—apart from small displacements due to the gyroscopic inertia couple Q—so that the very small friction forces parallel to V_c which are required to overcome "rolling resistance," will lie in one cross section of the roller and will be equal and opposite.

Section IX.—Now, for constructions according to the above relations of the dimensions, the roller may at very high constant shaft speeds exert a pressure force against the flange due to centrifugal force—as seen from equation (24).

In order that a friction force parallel to V_c due to such a pressure against the flange—which for constructions according to the relations of the dimensions is useful during changes of the shaft speed as it produces the angular acceleration of the roller about axis A_2C required for true rolling—should not exist at constant shaft speed, a construction, as shown in Fig. 6, is proposed.

In this construction the resultant of the pressure forces between the flange and the roller end is made to coincide with the generating line of the fixed race, which is the instantaneous axis of motion of the roller.

This is achieved by making the surface areas in contact between the flange and the roller end equal above and below the instantaneous axis of motion, and in this way the points on the roller head in contact with the flange below the instantaneous axis of motion will have their velocity relative to the flange

*[16] A "steady state" is meant to denote a state in which the mean value of the shaft speed is constant, *i.e.*, actually the angular velocity of the shaft varies about this mean value in a cycle. Thus, as already shown, the forces arising due to the changes of the shaft speed for a construction according to the relations of the dimensions will be useful—whereas for another construction they would produce skewing.

in the direction opposite to V_c, and therefore will have the sliding friction force in the direction of V_c, while points on the roller head in contact with the flange below the instantaneous axis of motion will have their velocity relative to the flange in the direction of V_c, and therefore the sliding friction force from this area will be in the direction opposite to V_c.

Thus the friction forces from the equal surface areas of contact will be equal and opposite and will give no resultant in the direction of V_c.

In the case of this type of construction, taking into account the fact that the friction force from the flange on the roller will always be zero in the direction parallel to V_c, the general equations for the relations of the dimensions will reduce to the following equations :—

$$I_2=\frac{Mr_m}{\tan\alpha_0}(l_{mc}-kr_m\tan\alpha_0) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (22b)$$

$$l_{mc}=\frac{1+k}{2(1-k)}\left(2\tan\alpha_0+\frac{C_f\sin\alpha_2}{N_1}\right)H-r_m\tan\alpha_0. \quad . \quad . \quad . \quad . \quad (19b)$$

$$\eta=\frac{1+k}{2(1-k)}\left(2\tan\alpha_0+\frac{C_f\sin\alpha_2}{N}\right)H-\tan\alpha_0(2r_m+r_{a1})\geqq 0 \quad . \quad . \quad (23b)$$

It is seen that the equations derived from the dynamic analysis, which give the relations of the dimensions for taper roller bearings to produce true rolling of the rollers at variable and constant shaft speed, require a certain displacement of the centre of gravity of the roller relative to the centre of gravity of its conical rolling surface.

Such positions of the centre of gravity of the roller must be between the cross section containing the centre of gravity of the generating line of the conical rolling surface and the inner (nearer to the shaft) end cross section of this rolling surface.

ANGULAR CONTACT BALL BEARINGS

General Remarks.

The relations of the dimensions as function of the conditions and the alterations of the constructions accordingly will be derived for angular contact ball bearings, such that true rolling of a ball at the point of action of the resultant of the pressure forces in the contact area between the ball and the fixed race track should be obtained during changes of the shaft speed as well as during a steady state.

To this end, the angle between the axis of the general angular velocity of the ball and the axis of its rolling angular velocity will first be determined as function of the dimensions of the bearing under the condition such that true rolling of the ball at the point of action of the resultant of the pressure forces in the contact area between the ball and the stationary race track should exist.

The relations of the dimensions as function of the conditions will then be developed, such that during changes of the shaft speed as well as during a state of constant shaft speed, the above mentioned angle between the axis of the general angular velocity and the rolling axis of the ball should be produced. Thus, relations of the dimensions must be determined such that the couples on the ball arising during changes of the shaft speed—which in a bearing where such relations of the dimensions are not present completely prevent rolling, *i.e.*, produce spin, etc.—should, due to these relations, produce such an angular acceleration and thus general angular velocity of the ball that the axes of the angular acceleration and of the general angular velocity coincide and the angle these two coincident axes form with the rolling axis is that angle, at which true rolling of the ball is produced at the point of action of the resultant of the pressure forces between the ball and the fixed race.

Further development of these relations of the dimensions leads to the determination of the conformity (radii and positions of centres of track-grooves in the rotating and fixed races, etc., determining the areas of contact between balls and tracks under load), such that there are eliminated the harmful effects of the couples which exist in the steady state due to the fact that for a ball making contact with its track over an area "true rolling" can only consist of true rolling at two points at the most within this contact-area.

Further, to take into account the fact that due to a radial load on a bearing, a ball will in different positions in a bearing be subject to different loads, constructions of double row bearings will be given which provide that the conditions to produce true rolling of a ball should be obtained for all positions of a ball in a bearing built according to the relations of the dimensions.

Section I.

Assuming that in an angular contact bearing, as indicated in Fig. 7, the resultant of the pressure forces in the contact area between the fixed race and the ball, acting at point A_2, makes an angle $90°-\alpha_2$ with the bearing (Y) axis, then the axis of the rolling angular velocity of the ball will have an angle α_2 with the bearing (Y) axis for true rolling of the ball at point A_2.

If ϕ is the angle between the axis of the general angular velocity ω of the ball and the rolling axis (as shown), then $\omega\cos\phi$ will be the angular velocity of the ball about the rolling axis, while $\omega\sin\phi$ will be the angular velocity of the ball about the axis A_2C through the point of action of the resultant and the centre of gravity.

If the angular velocity of the centre of gravity of the ball about the Y axis is denoted by ω_c, then ω_c will also be the angular velocity of the velocity vector V_c of the centre of gravity about the axis parallel to the Y axis through the centre of gravity, and thus $\omega\sin\phi$ will be the angular velocity of the vector V_c about the axis A_2C.

Now, for true rolling of the ball at the point of action A_2 of the resultant of the pressure forces between the ball and the fixed track, the velocity vector V_c must always lie in the same plane of the ball containing points A_2 and C (as has been already proved), thus the velocity vector V_c of the centre of gravity must lie in the plane normal to the axis of the rolling angular velocity of the ball, where this rolling axis has no relative motion with respect to the ball, and therefore the angular velocity of the ball and the angular velocity of the velocity vector V_c about the axis A_2C must be equal.

Thus
$$\omega\sin\phi=\omega_c\sin\alpha_2 \quad . \quad (1)$$

As now the angular velocity of the ball about the rolling axis is $\omega\cos\phi$, and for true rolling of the ball on the fixed track at point A_2, the relative velocity between the ball and the fixed track at A_2 must be zero, *i.e.*, A_2 must be the instantaneous centre of motion, therefore

$$V_{a2}=V_c-r\,\omega\cos\phi=0 \quad . \quad (2)$$

where r is the radius of the ball.

As the velocity of the centre of gravity
$$V_c=R_c\cdot\omega_c,$$
where R_c is the radius of the circle described by the centre of gravity of the ball about the bearing (Y) axis, from equation (1)

$$V_c=R_c\omega\,\frac{\sin\phi}{\sin\alpha_2} \quad . \quad . \quad (3)$$

From equations (2) and (3)

$$R_c=r\,\frac{\sin\alpha_2}{\tan\phi} \quad . \quad . \quad (4)$$

Thus relations of the dimensions as function of the angle ϕ, as required for true rolling of the ball at the point of action A_2 of the resultant of the pressure forces between the fixed track and ball, have been obtained.

If now point A_2 is connected with the point of intersection 0 of the axis of the rolling angular velocity ω of the ball with the bearing (Y) axis, and angle A_20C is

denoted by α_0, then it will be seen from the geometry of Fig. 7 that

$$R_c = r\frac{\sin\alpha_2}{\tan\alpha_0} \quad . \quad . \quad (5)$$

From equations (4) and (5) then

$$\tan\phi = \tan\alpha_0 . \quad . \quad (6)$$

This means that for true rolling of the ball at point A_2, the angle ϕ between the axis of the general angular velocity and the rolling axis of the ball must be equal to the angle α_0 between the rolling axis and the line $A_2 0$ connecting the point of action A_2 of the resultant between the fixed track and the ball with the point of intersection 0 of the rolling axis with the Y axis, *i.e.*, that the axis of the general angular velocity of the ball must for true rolling be parallel to the line $A_2 0$.

Section II.

The next stage is to determine the relations of the dimensions for which this angle ϕ will become equal to angle α_0, and in this way also to obtain angle α_0 as function of the conditions (speed, loading, etc.) : Relations of the dimensions have to be determined such that the axis of the angular acceleration of the ball (producing the general angular velocity) should coincide with the axis of the general angular velocity of the ball as required for true rolling at point A_2.

As the angular acceleration of the ball and the acceleration of its centre of gravity are functions of the forces on the ball in the direction parallel to the velocity vector V_c of its centre of gravity, the points of action of these forces have to be taken into account and according to the effects of these forces the relations of the dimensions have to be determined so that true rolling at the point of action A_2 of the resultant between the fixed race and the ball should be obtained during changes of the shaft speed as well as during a state of constant shaft speed.

These relations of the dimensions will be derived for two cases :

Case 1.—The centrifugal force taken up on rotating race track only (*i.e.*, the outer race rotates).

Case 2.—The centrifugal force taken up on the fixed race track only (*i.e.*, the inner race rotates).

For the construction in which the centrifugal force C_f on the ball is taken up on the rotating race track (Fig. 7), denote the angle which the resultant of the pressure forces between fixed race and ball makes with the Y axis by $(90° - \alpha_2)$. Then

$$\cot\alpha_2 = \frac{w}{t} \quad . \quad . \quad (7)$$

where w is the component in the radial direction, and t is the component in the axial direction of the pressure force between the fixed track and the ball.

Due to the centrifugal force C_f being taken up on the rotating track, the resultant of the pressure forces between the rotating track and the ball will make angle $(90° - \alpha_1)$ with the Y axis, where, as seen from the triangle of forces in Fig. 7,

$$\cot\alpha_1 = \frac{w + C_f}{t} = \frac{w}{t} + \frac{C_f}{t} = \cot\alpha_2 + \frac{C_f}{t} \quad (8)$$

From this

$$\sin(\alpha_2 - \alpha_1) = \frac{C_f}{t}\sin\alpha_2 . \sin\alpha_1 . \quad (8a)$$

where $(\alpha_2 - \alpha_1)$ is the angle between the resultants of the pressure forces between the ball and the fixed and rotating tracks respectively.

Now, let P_1 be the friction force in the direction parallel to V_c exerted by the rotating track on the ball during changes of the shaft speed (exclusive of the friction force required to overcome rolling resistance) acting at point A_1 a normal distance $r\cos(\alpha_2 - \alpha_1)$ from the rolling axis (see Fig. 7).

Let P_2 be the friction force parallel to V_c exerted by the fixed track on the ball during changes of the shaft speed (exclusive of the friction force required to overcome rolling resistance) acting at A_2 a normal distance r from the rolling axis.

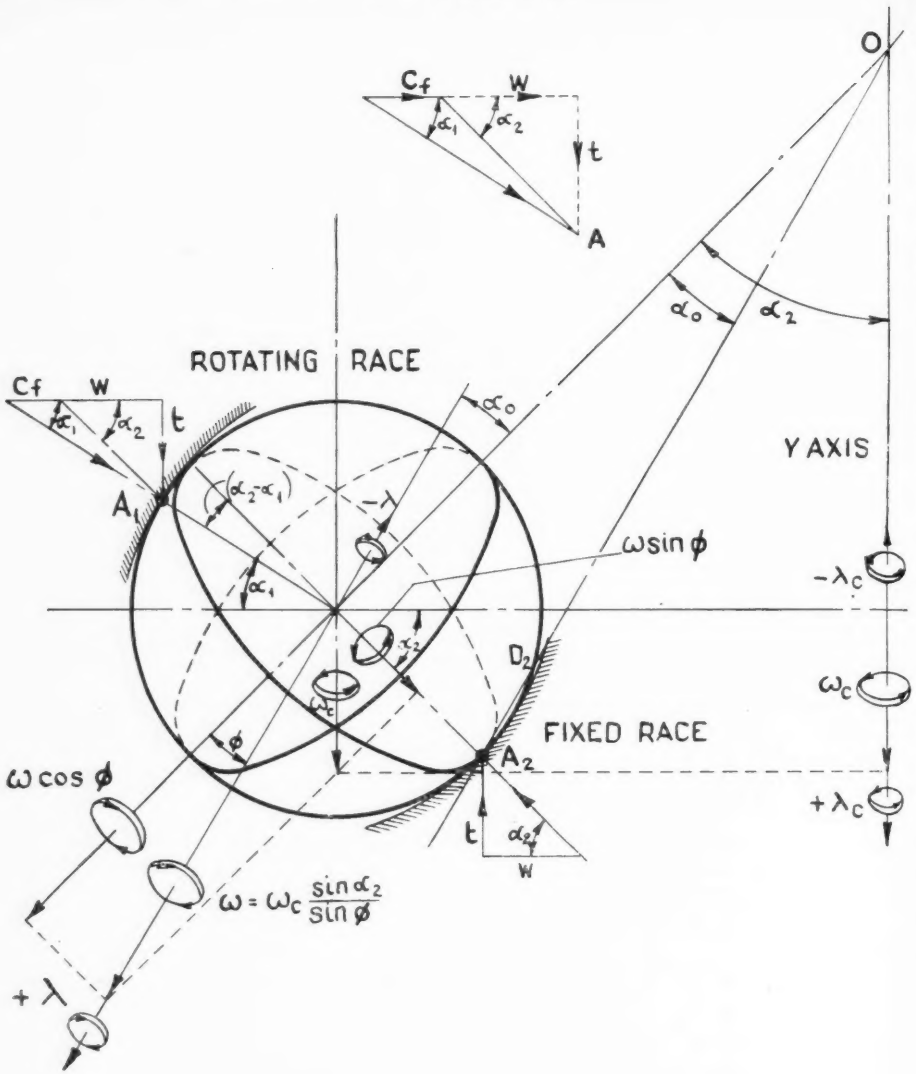


FIG. 7

As already generally proved (in the section on taper roller bearings), the relation between the driving force P_1 from the rotating track and the friction force P_2 from the fixed track must for true rolling at point A_2 during changes of the velocity V_c of the centre of gravity of the ball be

$$\frac{P_1}{P_2} = \frac{1-k/\cos\varphi}{1+k} \cdot \cos\varphi \quad . \qquad (9)$$

In this case the normal distance of the driving force P_1 from the rolling axis is $r\cos\varphi = r\cos(\alpha_2 - \alpha_1)$, and for a ball the radius of gyration $k=0.4$, so that in general

$\cos(\alpha_2-\alpha_1)>k$ and Case 1 of the three cases previously treated *17 applies.

Thus

$$\frac{P_1}{P_2}=\frac{1-k/\cos(\alpha_2-\alpha_1)}{1+k}.\cos(\alpha_2-\alpha_1) \quad (9a)$$

where forces P_1 and P_2 must be in the same direction as the acceleration of the centre of gravity.

Now, the friction force P_1 together with its dynamic reaction at the centre of gravity of the ball will form the couple $P_1 r\cos(\alpha_2-\alpha_1)$ in the plane normal to the rolling axis, while the friction force P_2 from the fixed track together with the dynamic force it produces at the centre of gravity will form the couple $P_2 r$ in the plane normal to the rolling axis.

The resultant couple in the plane normal to the rolling axis

$$P_1 r\cos(\alpha_2-\alpha_1)\left[1-\frac{P_2}{P_1\cos(\alpha_2-\alpha_1)}\right]=I\lambda_1 \quad (10)$$

will produce the angular acceleration λ_1 about the rolling axis, while the moment due to the friction force P_1 about the axis A_2C normal to the rolling axis

$$P_1 r\sin(\alpha_2-\alpha_1)=I\lambda_2 \quad . \quad (11)$$

will produce the angular acceleration λ_2 about the axis A_2C normal to the rolling axis.

From the condition for true rolling, namely, that the axis of the angular acceleration λ of the ball should coincide with the axis of the general angular velocity as required for true rolling at point A_2,—this axis, as already proved, making angle $\phi=\alpha_0$ with the rolling axis—it follows that

$$\left.\begin{aligned}\lambda_1&=\lambda\cos\alpha_0\\ \lambda_2&=\lambda\sin\alpha_0\end{aligned}\right\}. \quad . \quad (12)$$

By substituting equation (12) in equations (10) and (11) and dividing

$$\frac{P_1 r\sin(\alpha_2-\alpha_1)}{P_1 r\cos(\alpha_2-\alpha_1)\left[1-P_2/P_1\cos(\alpha_2-\alpha_1)\right]}=\tan\alpha_0 \quad (13)$$

Substituting for P_1/P_2 from equation (9*a*),

$$\tan\alpha_0=\tan(\alpha_2-\alpha_1)\frac{1}{1-\dfrac{1-k/\cos(\alpha_2-\alpha_1)}{1+k}}$$

$$=\tan(\alpha_2-\alpha_1)\frac{1+k}{k\left(1+\dfrac{1}{\cos(\alpha_2-\alpha_1)}\right)}=\frac{\tan(\alpha_2-\alpha_1)\cos(\alpha_2-\alpha_1)}{\cos(\alpha_2-\alpha_1)+1)}.\frac{(1+k)}{k}$$

$$=\frac{\sin(\alpha_2-\alpha_1)}{2\cos^2\left(\dfrac{\alpha_2-\alpha_1}{2}\right)}.\frac{(1+k)}{k}=\frac{1+k}{k}.\tan\left(\frac{\alpha_2-\alpha_1}{2}\right)$$

so that

$$\tan\alpha_0=\frac{1+k}{k}\tan\left(\frac{\alpha_2-\alpha_1}{2}\right) \quad . \quad . \quad . \quad . \quad . \quad (14)$$

From equation (14), giving the angle α_0 between the line $A_2 0$ and the rolling axis, it is seen that this angle is only a function of the difference $(\alpha_2-\alpha_1)$, but not of angle α_2 separately or of angle α_1 separately.

From equation (14), in conjunction with equation (5) from the geometry of Fig. 7,

$$R_c=\frac{r\sin\alpha_2}{\tan\alpha_0} \quad . \quad . \quad (5),$$

the relation of the dimensions

$$R=r\frac{\sin\alpha_2}{\tan\left(\dfrac{(\alpha_2-\alpha_1)}{2}\right)}.\frac{k}{(1+k)} \quad (15)$$

is obtained.

Equation (15) gives the relations of the

*17 The three cases quoted are proved in the section on "Taper Roller Bearings."

dimensions for Case 1 of the rotating race only of the angular contact bearing taking up the centrifugal force (outer race rotating).

A similar proof applies to Case 2 of the fixed race only taking up the centrifugal force (inner race rotating), as shown in Fig. 7(*a*).

In this case, for true rolling at the point of action A_1 of the resultant of the pressure forces between the fixed race and the ball, rolling will be at point A_1 so that now the rolling axis will make angle α_1 with the bearing (Y) axis (using the same symbols as in Fig. 7 and in the preceding proof).

As seen with reference to Fig. 7(*a*), for the same direction of rotation of the shaft, the component of the general angular velocity in the direction of the rolling axis will be in the

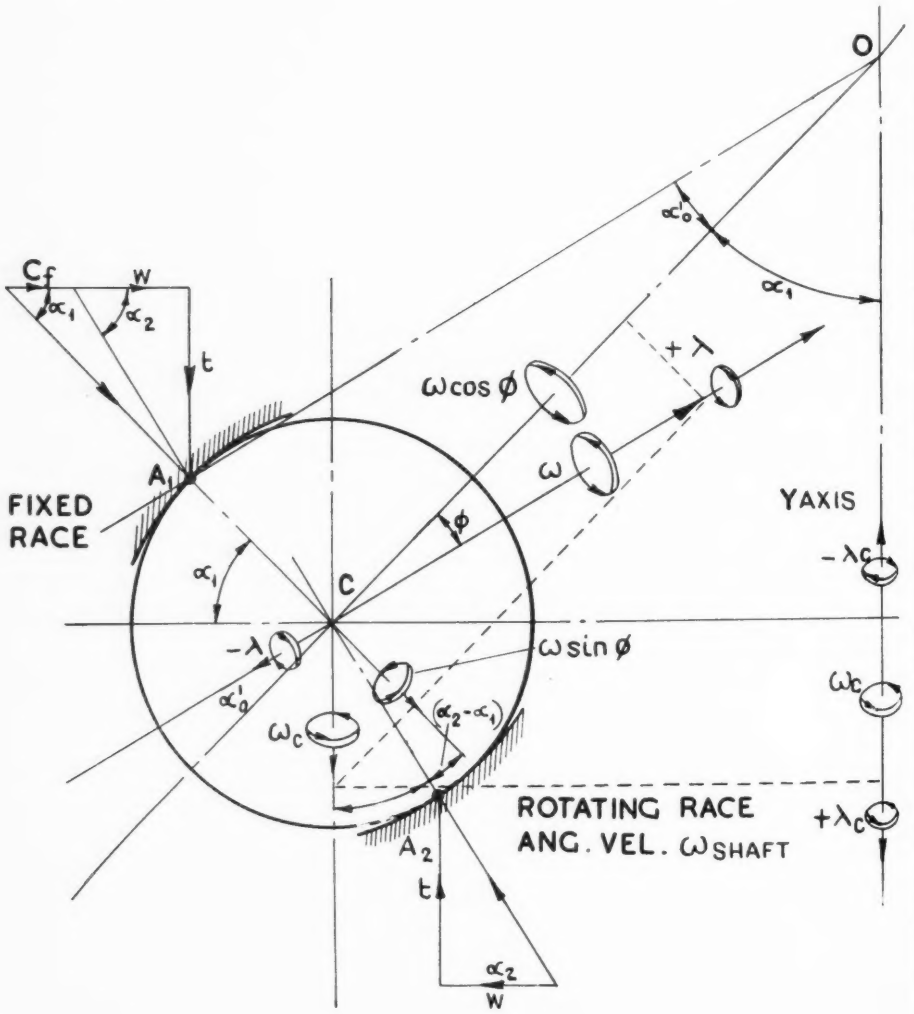


FIG. 7A

opposite direction along the rolling axis as in Case 1, while the component of the general angular velocity of the ball in the direction A_1C must again for true rolling at A_1 be equal in direction and magnitude to the component along A_1C of the angular velocity ω_c.

By the proof as developed for Case 1, for angle ϕ

$$\tan\phi = \tan\alpha^1{}_0$$

for true rolling at A, in other words, the axis of the general angular velocity of the ball must for true rolling at point A_1 be parallel to the line $A_1 0$ connecting the point of action A_1 of the resultant between the fixed track and the ball with the point of intersection 0 of the rolling axis with the bearing (Y) axis.

In this Case 2, during changes of the shaft speed, the point of action of the driving force P_2 parallel to V_c will be A_2, and the couple due to the force P_2 in the plane normal to axis A_1C will be

$$P_2 r \sin(\alpha_2 - \alpha_1) = I\lambda_2 = I\lambda\sin\alpha^1{}_0 \qquad (11a)$$

producing the component $\lambda_2 = \lambda\sin\alpha^1{}_0$ of the angular acceleration λ of the ball as required for true rolling at A_1, while the couple in the plane normal to the rolling axis due to forces P_2 and P_1 will be

$$P_2 r\cos(\alpha_2 - \alpha_1)\left[1 - \frac{P_1}{P_2\cos(\alpha_2 - \alpha_1)}\right] = I\lambda_1 = I\lambda\cos\alpha_2. \qquad (10a)$$

producing the component $\lambda_1 = \lambda\cos(\alpha_2 - \alpha_1)$ of the required angular acceleration λ of the ball.

By division of the equations (11*a*) and (10*a*) and substitution of the appropriate relation between driving force P_2 from the rotating track and friction force P_1 from the fixed track for true rolling according to equation (9*a*), and from the geometry of Fig. 7(*a*), the following relation of the dimensions is obtained :—

$$R_c = r\frac{\sin\alpha_1}{\tan\left(\frac{\alpha_2 - \alpha_1}{2}\right)}\cdot\frac{k}{1+k} \quad . \qquad (15a)$$

Thus, for Case 1 of the centrifugal force being taken up on the rotating track (outer race rotating), the relation of the dimensions is

$$R_c = r\frac{\sin\alpha_2}{\tan\left(\frac{\alpha_2 - \alpha_1}{2}\right)}\cdot\frac{k}{1+k} \quad . \qquad (15)$$

while for Case 2 of the centrifugal force being taken up on the stationary track (inner ring rotates), the relation of the dimensions is

$$R = r\frac{\sin\alpha_1}{\tan\left(\frac{\alpha_2 - \alpha_1}{2}\right)}\cdot\frac{k}{1+k} \quad . \qquad (15a)$$

Thus the relation R/r is a function of the angles α_2 and α_1, the angles between the resultants of the pressure forces between the races and the ball, and of $k = \frac{I}{Mr^2}$ where $k = 0.4$ for a solid ball.

The expressions for the angles α_2 and α_1 in the form required for the relations of the dimensions is obtained from the development of equation (8)

$$\frac{C_f}{t} = \cot\alpha_2 - \cot\alpha_1 = \frac{\sin(\alpha_2 - \alpha_1)}{\sin\alpha_1 . \sin\alpha_2} \quad . \quad . \quad . \quad . \quad . \quad . \qquad (8)$$

From this

$$\frac{\sin(\alpha_2 - \alpha_1)}{\sin\alpha_2} = \frac{C_f}{t}\sin\alpha_1 = \frac{C_f}{t}\sin(\alpha_1 - \alpha_2 + \alpha_2) = \frac{C_f}{t}[\sin(\alpha_1 - \alpha_2)\cos\alpha_2 + \cos(\alpha_2 - \alpha_1)\sin\alpha_2]$$
$$= \frac{C_f}{t}\cos(\alpha_2 - \alpha_1)\sin\alpha_2 - \tan(\alpha_2 - \alpha_1).\cos\alpha_2].$$

From this

$$\frac{\tan(\alpha_2 - \alpha_1)}{\sin\alpha_2} = \frac{C_f}{t}\sin\alpha_2 - \frac{C_f}{t}\frac{\tan(\alpha_2 - \alpha_1)}{\sin\alpha_2}.\sin\alpha_2.\cos\alpha_2,$$

and therefore

$$\frac{\tan(\alpha_2-\alpha_1)}{\sin\alpha_2}=\frac{(C_f/t).\sin\alpha_2}{1+(C_f\,t)\sin\alpha_2.\cos\alpha_2}$$

and because

$$\frac{\tan(\alpha_2-\alpha_1)}{\sin(\alpha_2)}=\frac{2\tan\left(\frac{\alpha_2-\alpha_1}{2}\right)}{\left[1-\tan^2\frac{(\alpha_2-\alpha_1)}{2}\right]\sin\alpha_2}=\frac{\frac{C_f}{t}.\sin\alpha_2}{\left[1+\frac{C_f}{t}\sin\alpha_2\ \cos\alpha_2\right]}$$

therefore

$$\frac{\sin\alpha_2}{\tan\left(\frac{\alpha_2-\alpha_1}{2}\right)}=\frac{2t}{C_f\sin\alpha_2}+2\cos\alpha_2+\tan\left(\frac{\alpha_2-\alpha_1}{2}\right)\sin\alpha_2$$

so that

$$\frac{\sin\alpha_2}{\tan\left(\frac{\alpha_2-\alpha_1}{2}\right)}\approx\frac{2}{C_f}\cdot\frac{t}{\sin\alpha_2}+2\cos\alpha_2 \qquad . \quad . \quad . \quad . \quad . \quad (16)$$

By substituting $-\alpha_1$ for α_2, and $-\alpha_2$ for α_1, in (16)

$$\frac{\sin\alpha_1}{\tan\left(\frac{\alpha_2-\alpha_1}{2}\right)}\approx\frac{2}{C_f}\cdot\frac{t}{\sin\alpha_1}-2\cos\alpha_1 \qquad . \quad . \quad . \quad . \quad . \quad (16a)$$

and thus the expressions as required in the relations of the dimensions (15) and (15*a*) for Cases 1 and 2 are obtained.

Thus for Case 1—Fig. 7—

$$R=r\left(\frac{2}{C_f}\cdot\frac{t}{\sin\alpha_2}+2\cos\alpha_2\right)\frac{k}{1+k} \qquad . \quad . \quad . \quad . \quad . \quad (15b)$$

and for Case 2—Fig. 7(*a*)—

$$R=r\left(\frac{2}{C_f}\cdot\frac{t}{\sin\alpha_1}-2\cos\alpha_1\right)\frac{k}{1+k} \qquad . \quad . \quad . \quad . \quad . \quad (15c)$$

The meaning of these expressions will emerge in the following treatment of the loading conditions due to a radial load W for a ball in different positions in the bearing, in connection with the construction required to produce the conditions for true rolling.

Section III.

As the forces at A_1 and A_2 are the resultants of the pressure forces between a ball and the races, this means that the contact areas of this ball with the races must be symmetrical about the vectors of the resultant pressure forces, and thus in both cases the axes of symmetry of the track grooves in the rotating and stationary races respectively must be made such that they are not in the same straight line, but include an angle $(\alpha_2-\alpha_1)$, as shown in Fig. 9, so that the triangles of forces as shown in Fig. 7 will apply.

Now, for a bearing with no radial load, but only axial load, if such track grooves are constructed that their axes of symmetry coincide with the resultant forces on the ball at the time when the axial load is a maximum, then the triangle of force will, at times when the axial load is smaller, but the centrifugal force C_f is the same, produce the axial and radial reactions t_r and a_r on each ball, where

$$t_r = t_{max}$$
$$w_r = w_{max}.$$

Now, it is known that a radial load W on a bearing will impose on a ball in its line of action a radial load of $4.36 \frac{W}{z}$ *18 (where z is the number of balls in the bearing), while on the ball in the diametrically opposite position it will cause no radial load.

Thus there will be a difference in the radial pressure forces on the most heavily and least loaded balls in the bearing of

$$w_{max} - w_{min} = 4.36 \frac{W}{z} \quad . \quad (17)$$

so that due to the radial load W on the bearing the axial components of the resultant pressure forces on these balls will differ by

$$t_{max} - t_{min} = 4.36 \frac{W}{z} . \tan \alpha \quad . \quad (18)$$

If now, in the case of a radial load W acting together with axial load, a bearing is constructed such that the axes of symmetry of the track grooves in the stationary and rotating races coincide with the directions of the resultants of the pressure forces to which the ball becomes subject in the position for maximum load (according to the triangle of forces as shown in Fig. 7), then the directions of the resultant pressure forces on the ball in positions other than that for maximum load will be different (the triangles of forces will for different positions of the ball in the bearing differ from the triangle of forces for the position of maximum load). In other words, the resultants of the pressure forces will not coincide with the axes of symmetry of the track grooves, because the radial load on a ball in the position of maximum loading must be larger by $4.36\ W/z$ than the radial pressure force on the ball in the position diametrically opposite.

Now four different cases will be treated :—

(1) The axial load T negligibly small compared to radial load W on the bearing.

(2) Where both radial load W and axial load T have to be taken into account.

(3) Where the radial load W and the axial load T are very large.

(4) Where the axial load T is large and W is negligibly small.

Case 1.—For the case of axial load negligibly small compared with the radial load W, the directions of the resultant pressure forces on a ball in position other than that of maximum loading will not coincide with the axes of symmetry of the track grooves, which are made to coincide with the directions of the resultant pressure forces on the ball in the position of maximum loading (the triangles of forces will, for different positions of the ball in the bearing, differ from the triangle of forces for the maximum load position), because the radial load on a ball in the position of maximum load must be larger by $4.36\ W/z$ than the radial reaction on the ball in the diametrically opposite position.

Now, in the case of a double row bearing, as usual for angular contact bearings, the radial load W is distributed on two rows of balls, and for each row there will thus be a difference between the maximum load and the radial reaction on the least loaded ball of

$$w_{max} - w_{min} = 2{\cdot}18 W/z,$$

where z is the number of balls per row, and thus due to the radial load W on the bearing a difference in the maximum and minimum axial reactions of

$$t_{max} - t_{min} = 2{\cdot}18\ W/z . \tan \alpha.$$

In order, now, to reduce these differences between the reactions in the axial direction to negligibly small quantities, a construction of double row bearing is proposed for which the positions for maximum loading on a ball due to the radial load W in one row, should be diametrically opposite to the position for maximum loading on a ball due to W in the second row ; in other words, this position in the one row should be rotated by 180° relative to this position in the other row. This is shown in Fig. 8.

*18 According to Stribeck.

This is achieved by constructions of double row bearing in which—in contrast to the usual double row bearing where one ring has the inner races for both rows of balls and the other ring the outer races for both rows of balls—each bearing ring has on one side the inner race for one row of balls and on the other side the outer race for the second row of balls, as is shown in the examples on Figs. 8 and 8(*a*).

The angle of contact α should be $60° < \alpha < 90°$ where by angle of contact is meant the angle between the axis of symmetry of a track groove and the radial direction. The stationary ring of the bearing should be on a spherical mounting, as shown.

Now, in Fig. 8, for a radial load W as shown, the position for maximum load in row 1 will be at A and the position for maximum load in row 2 will be at C.

Each row of balls carries half the radial

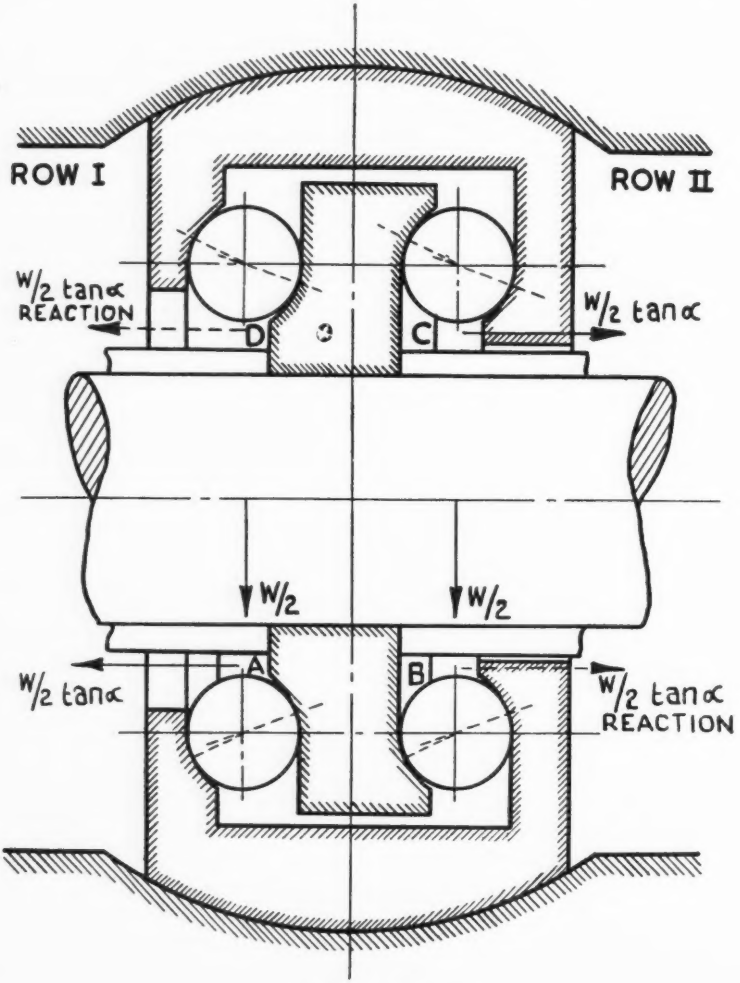


FIG. 8

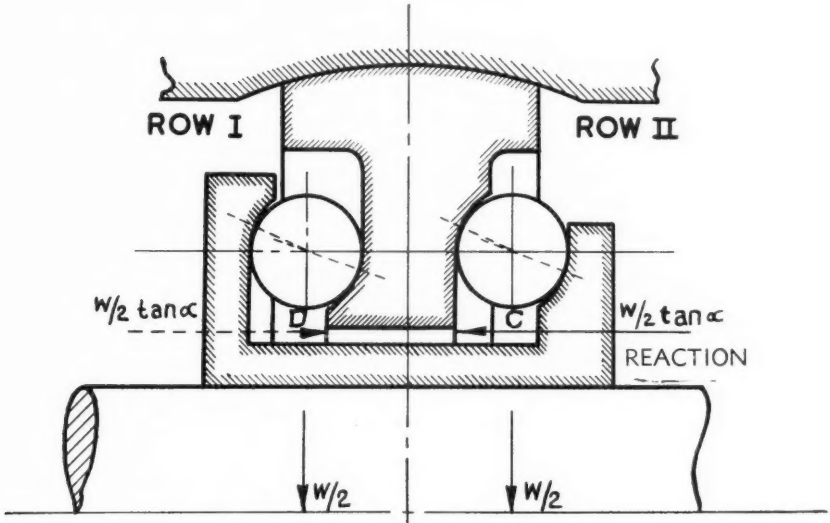


FIG. 8.A

load W, and will, as will be shown in the following, cause reactions in the axial direction of the balls in the other row :—

That half of the balls in row 1, which carries the radial load $W/2$, will exert on the stationary ring the total axial force $W/2.\tan\alpha$, with the line of action through A as indicated on Fig 8 ; while the balls in row 2 which carry $W/2$, will exert on the stationary ring their total axial force $W/2.\tan\alpha$ with the line of action through C, as indicated in heavy lines on Fig. 8.

These two axial forces form a couple on the stationary ring, which is mounted so that it could swivel in the housing about centre O, and therefore this couple produces an opposite couple through causing the axial reactions $W/2.\tan\alpha$ with lines of action through B and D, as shown in dotted lines, *i.e.*, this couple causes the axial reactions $W/2.\tan\alpha$ on those balls in each row which do not take part in carrying the radial load $W/2$ on that row.

Thus—in contrast to orthodox double row bearings, in which no reactions in the axial direction at all are caused in that half of the balls in one row, which is not radially loaded by W—in this type of construction this half of the balls in row 1 which carried the radial load $W/2$ on this row, will cause an axial reaction $W/2.\tan\alpha$ on that half of the balls in row 2, which does not take part in carrying radial load $W/2$ on row 2 ; while this half of the balls in row 2, which carries the radial load $W/2$, will cause the axial reaction $W/2.\tan\alpha$ on that half of the balls in row 1, which does not take part in carrying the radial load $W/2$ on row 1.

In this way, in every row, half of the balls will be subject to an axial pressure $W/2.\tan\alpha$ because they carry the radial load W/2, while the second half of the balls in this row will be subject to an axial reaction $W/2.\tan\alpha$, and in this way the balls in each row will be subject to a total axial pressure $W.\tan\alpha$, whose resultant passes through the centre

of the row.

In this way, because the resultant of the axial pressure forces on each row passes through its centre, the differences of the axial pressure forces on balls in different positions in the bearing are made negligibly small, but this is only for the constructions in which the angle of contact $60° < \alpha < 90°$.

This will be represented exactly in a further paper, in connection with the elastic deformations which occur.*19

In this way the triangles of forces will become approximately the same for balls in all positions in the bearing.*20

Case 2.—In the case where the axial load is not negligible, but not too large, the stationary bearing ring in the construction shown in Fig. 8 should have an axial preload which is equal to the axial load on the bearing, so that one row of balls will take up an axial load of $2T + W.\tan\alpha$, while the other row of balls will take up at least the axial reaction of $W.\tan\alpha$.

For a case of the external axial load remaining in one direction only, the balls and the track grooves in each row should have such radii that the contact areas in the row of balls which has to take up the axial load $2T + W.\tan\alpha$, and the contact areas between balls and races in the row taking up the axial reaction $W.\tan\alpha$ should be in the proportion.

$$\frac{S_I}{S_{II}} = \frac{2T + W.\tan\alpha}{W\tan\alpha}$$

so that the mean pressure intensities should be about equal.

The relations of the dimensions will be determined accordingly for each row of balls.

Case 3.—When the axial load as well as the radial load are large, a double row bearing of the type proposed should be used for the carrying of the radial load only, while the large axial load is carried by a special thrust bearing.

Case 4.—For a bearing on which the radial load W is negligibly small compared with the axial load T, the triangle of forces on the ball will be practically the same for all positions of the ball in the bearing. At times when the axial load is smaller, but the centrifugal force C_f is the same, the same axial and radial reactions t_r and w_r on a ball will be produced as in the case of maximum axial load—t_r and w_r will be equal to the maximum axial and radial forces on the balls—because the axes of symmetry of the track grooves are made such as to coincide with the directions of the resultant pressure forces on the balls for maximum load.

Section IV.

In general, it would be advantageous that the cage, which otherwise only acts as a ball separator, should have component parts on its inner ring which exert an outward pressure force on the ball in the direction of its rolling axis, so that balls which are somewhat smaller than others should also take part in the carrying of the load—such a construction is indicated in Fig. 9*a*.

The influence of the cage, which rotates with the same angular velocity ω_c about the shaft as the centre of gravity of a ball, will be in harmony with the influence on the ball of the angular contact couples from the races, for relations of the dimensions which produce true rolling, as emerges from the following consideration:

According to the general proof (given in "General Considerations"), all particles of the cage have the angular velocity ω_c about the axis through the centre of gravity of a ball parallel to the Y axis, and thus have the angular velocity $\omega_c.\sin\alpha_2$ about the axis A_2C, as shown in Fig. 7. If now the relations of the dimensions are such that the angular contact couples on the ball give it the angular velocity $\omega_c.\sin\alpha_2$ about the axis A_2C as required for true rolling at A_2,

*19 It is already clear without the exact proof that any radial displacement due to W, which for an angle of contact $\alpha < 45°$ will cause slackness, will for the angle of contact $60° < \alpha < 90°$ require a correspondingly larger displacement in the axial direction, which is therefore connected with an elastic deformation in the axial direction, etc.

*20 The bearing should have a small axial preload.

say, then there will be no relative velocity (in the plane normal to A_2C) between the cage and the ball at their point of contact on the rolling axis of the ball, and thus there will be no couple on the ball due to a friction force at this point of contact to disturb the true rolling.

As indicated in Fig. 9(*a*), the outward pressure on the ball in the direction of the rolling axis can be exerted by a suitably shaped centrifugal body mounted on the inner ring of the cage on a pin in the direction of the rolling axis of each ball, the body being mounted in such a way that it can rotate about the pin, and slide along it under centrifugal force, and thus exert the required pressure on the ball.

The relations of the dimensions for this case will be given in another paper.

Section V.

For a bearing constructed to the relations of the dimensions derived above, the forces P_1 and P_2 on a ball in the direction parallel to the velocity vector V_c of its centre of gravity, which arise during changes of the shaft speed, produce such angular acceleration and thus such general angular velocity of a ball, that the axes of the angular acceleration and the general angular velocity of the ball coincide and form that angle with the rolling axis, at which true rolling of the ball at the point of action of the resultant of the pressure forces between the fixed race and the ball is obtained.

In constructions where these relations of the dimensions are not adhered to, these same couples due to the forces P_1 and P_2 produce spin, etc., and completely prevent rolling.

Now, when the shaft speed is constant, the "rolling resistance" which exists even for true rolling, will produce small friction forces in A_1 and A_2 in the direction parallel to velocity vector V_c of the centre of gravity. In order to prevent these forces producing spin of the balls about the axis A_2C, the contact areas between balls and races will be determined, as follows:—

As shown on Figs. 9 and 9(*a*), the line connecting A_2 with O, the point of intersection of the rolling axis and the bearing (Y) axis, intersects the surface of the ball at point D_2. Thus, for the relations of the dimensions giving true rolling of the ball at point A_2, A_2D_2 will be the instantaneous axis of motion of the ball, *i.e.*, the ball will instantaneously rotate about axis A_2D_2 for true rolling at point A_2.

Therefore the points on the ball in contact with the fixed race below line D_2A_2 will have their velocity relative to the fixed race in the direction opposite to V_c, and therefore the sliding friction force will be in the direction of V_c, while the points on the ball in contact with the fixed race above line D_2A_2 will have their velocity relative to the fixed race in the direction of V_c, so that the sliding friction force will be in the direction opposite to V_c.

For this reason the track groove must be made symmetrical about A_2 with such a radius,*[21] that for the maximum resultant pressure designed for, the contact area B_2E_2 should be such that

$$B_2A_2 = A_2E_2 < A_2D_2,$$

for then the friction force in the area A_2B_2 in the direction of V_c and the friction force in the equal area A_2E_2 in the direction opposite to V_c will give no resultant friction force in the direction parallel to V_c—see Fig. 9(*a*).

As regards the couple formed by these friction forces in A_2B_2 and A_2E_2, this, as will be shown in the following, is in the opposite direction to the couple formed by the friction forces acting from the rotating race track on the ball, and these two couples will balance.

The arc A_2D_2 is given by angle $A_2\hat{C}D_2$ where, as can be seen from the geometry of Fig 9,

$$A_2\hat{C}D_2 = 2.\hat{\alpha}_9,$$

*[21] The radius of the track groove will be determined according to loading considerations from the deformation formulae of Heinrich Hertz, as given in the relevant literature.

angle $\hat{\alpha}_0$ being obtained from equation (14)

$$\tan\alpha_0=\tan\left(\frac{\alpha_2-\alpha_1}{2}\right)\frac{k+1}{k} \quad . \quad (14)$$

Now for the following reasons it is necessary that

$$A_2B_2=A_2E_2<A_2D_2.$$

(1) If $A_2B^1{}_2>A_2D_2$, the resultant friction force in the direction parallel to V_c acting from the fixed race on the ball would no longer be zero.

This is because then there would be in the direction opposite to V_c the friction force from the area A_2E_2 and also from the area $D_2B^1{}_2$ (see Fig. 9(*a*)), while in the direction of V_c there will only be the friction force from area A_2D_2, so that there will be a resultant friction force in the direction opposite to velocity vector V_c.

(2) $A_2B_2=A_2E_2<A_2D_2$ particularly for a bearing not of the types as shown in Figs. 8 and 8(*a*), because A_2 is then only the assumed point of action of the resultant of the pressure forces, while the actual point of action $A^1{}_2$ moves about point A_2, and thus, when $A^1{}_2$ comes between A_2 and D_2, a part of the contact area $A^1{}_2B^1{}_2$ would move beyond point D_2 (or beyond point $D^1{}_2$ corresponding to point $A^1{}_2$, see Fig 9(*a*)), and thus the resultant of the friction force in the direction parallel to V_c would no longer be zero.

Thus, taking into account these factors,

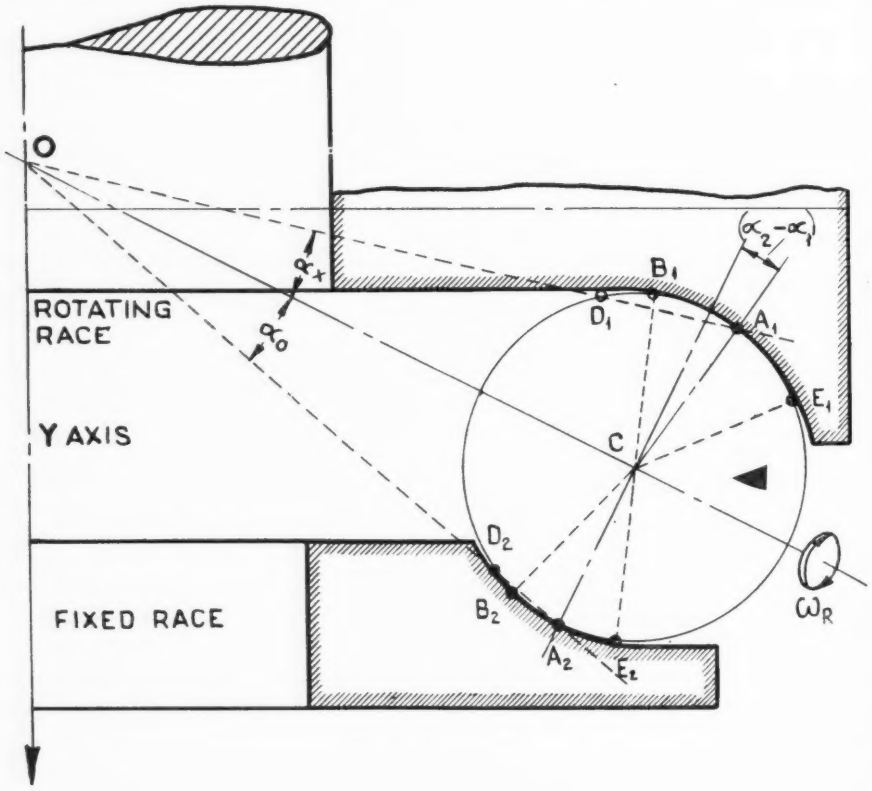


FIG. 9

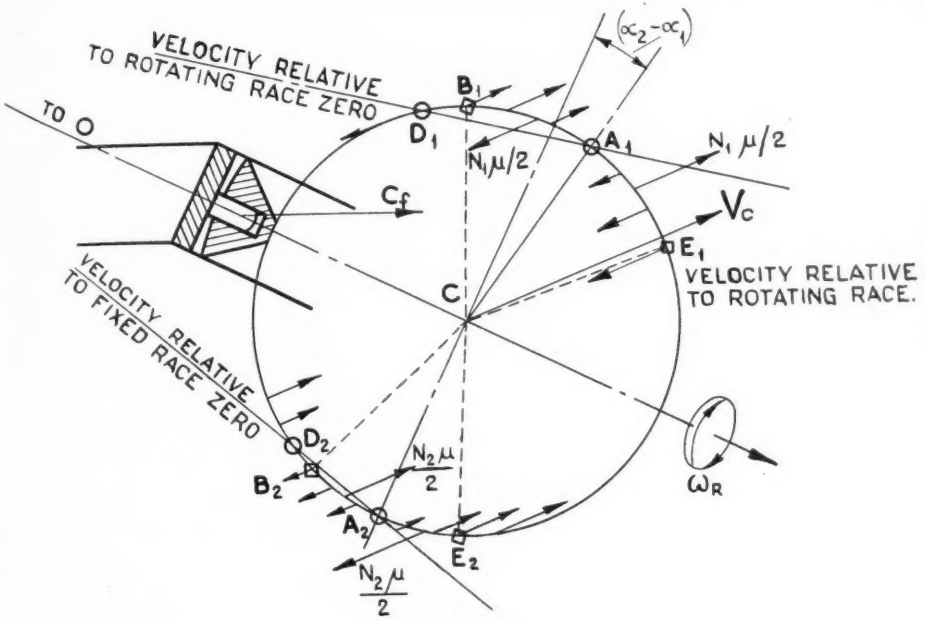


FIG. 9.A

the radius of the track groove must be so determined that

$$A_2B_2=A_2E_2\leqq A_2D_2.$$

As regards the contact area (conformity) between the rotating race and the ball, the following development arises :—

Let the line connecting the point of action A_1 of the resultant of the pressure force between ball and rotating race with the point of intersection O of the rolling axis with the Y axis intersect the surface of the ball at D_1. Because for a bearing according to the relations of the dimensions derived above, the relative velocity between the ball and the rotating race at point A_1 will be zero, tnerefore the line A_1O will be the locus of all points of the ball which have no relative velocity with respect to the rotating race. Therefore the points on the ball in contact with the rotating race above line A_1O will have their relative velocity with respect to the rotating race in the direction of V_c, and thus the sliding friction forces will be in the direction opposite to V_c, while the points on the ball in contact with the rotating race below line A_1O will have their relative velocity with respect to the rotating race in the direction opposite to V_c, so that the sliding friction force will be in the direction of V_c.

Therefore it is necessary to provide for such a radius of the track groove (whose axis of symmetry will make an angle $(\alpha_2-\alpha_1)$ with the axis of symmetry of the track groove in the fixed race), that the contact area B_1E_1 under load will be such that

$$A_1B_1=A_1E_1<A_1D_1,$$

so that the friction forces from the equal areas A_1B_1 and A_1E_1, being in opposite directions, should give no resultant friction force in the direction parallel to V_c.

Because the directions of the sliding friction forces in the areas A_1B_1 and A_1E_1 on the rotating race track are opposite to the directions of the sliding friction forces in the corresponding areas A_2B_2 and A_2E_2 on

the fixed race track, as has been shown above, the couples formed by the friction forces on the ball at the fixed and rotating races respectively will balance.

The same two reasons as apply for the contact area at the fixed race exist for making the contact area with the rotating race such that

$$A_1B_1=A_1E_1<A_1D_1.$$

The radii of the track grooves in the rotating and fixed races respectively must be calculated so that the contact areas B_2E_2 and B_1E_1 should not be equal, but should be in the relation

$$\frac{B_2E_2}{B_1E_1}=\frac{A_2\hat{C}D_2}{A_1\hat{C}D_1}.$$

From the geometry of Fig. 9

$$A_1\hat{C}D_1=2[\alpha_x+(\alpha_2-\alpha_1)]$$

and

$$\cot\alpha_x=\frac{\cot\alpha_0}{\cos(\alpha_2-\alpha_1)}+\tan(\alpha_2-\alpha_1)$$

angle α_0 having been determined from equation (**14**), which is

$$\tan\alpha_0=\tan\left(\frac{\alpha_2-\alpha_1}{2}\right)\frac{1+k}{k}.$$

THE SOCIETY'S AWARDS

The Society offers a number of valuable awards, most of them annually. Full particulars of the conditions attaching to these awards may be obtained on application to the Secretary.

Society's Gold Medal

This is the highest honour which the Society can confer for work of an outstanding or fundamental nature in aeronautics.

British Gold Medal for Aeronautics

The British Gold Medal for Aeronautics is awarded for an achievement leading to advancement in aeronautical science.

Simms Gold Medal

The Simms Gold Medal is awarded annually for the best paper read in any year before the Society on any science allied to aeronautics, e.g., meteorology, wireless telegraphy, instruments.

The George Taylor (of Australia) Gold Medal

The Taylor Gold Medal is awarded annually, at the discretion of the Council, for the most valuable paper submitted or read during the previous session.

Wakefield Gold Medal

The Wakefield Gold Medal is awarded annually to the designer of any invention or apparatus tending towards safety in flying, and is open to members or non-members.

Society's Silver Medal

The Society's Silver Medal is awarded, at the discretion of the Council, for some advance in aeronautical design.

British Silver Medal for Aeronautics

The British Silver Medal for Aeronautics is awarded for an achievement leading to advancement in aeronautical science.

Society's Bronze Medal

The Society's Bronze Medal is awarded, at the discretion of the Council, under the same conditions as those for the Silver Medal, but for some less important advance in aeronautical design.

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The Wilbur Wright Memorial Lecture is held annually, a premium of seventy-five pounds being awarded to the lecturer invited by the Council to deliver the lecture. The lecture is usually given alternately by an American and an Englishman, and is the most important aeronautical lecture of the year. It is delivered whenever possible, on the last Thursday in May of each year.

British Empire and Commonwealth Lecture

The Council of the Royal Aeronautical Society have completed the arrangements for the founding of a British Empire and Commonwealth Lecture.

The lecture, on any aeronautical subject approved by the Council, will be delivered annually in September in London, by a lecturer, chosen in alternate years from the British Dominions and Colonies and Great Britain.

The Council, by founding this British Empire and Commonwealth Lecture, are anxious to encourage new ideas and new points of view from all parts of the British Empire and Commonwealth, and to make the Lecture second only in importance to the Wilbur Wright Memorial Lecture.

The British Empire and Commonwealth Lecture will have a premium of £50 attached to it, and in the case of lecturers coming from the Dominions and Colonies an allowance will be paid towards the Lecturer's expenses.

R.38 Memorial Prize

The R.38 Memorial Prize is offered annually for the best paper received by the Society on some subject of a technical nature in the science of aeronautics, preference being given to papers which relate to airships. The prize is twenty-five guineas.

The Herbert Akroyd Stuart Lectures

Under the will of the late Mr. Herbert Akroyd Stuart a sum of £700 is held in trust by the Society for the offer of a prize every two years, for the best paper or lecture read or given before the Society dealing with the Origin and Development of Heavy-oil Aero Engines. The prize is open to members and non-members.

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The Edward Busk Memorial Prize is offered annually for the best paper received by the Society on some subject of a technical nature in connection with aeroplanes (including seaplanes). Its value is twenty guineas.

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The Pilcher Memorial Prize is offered annually, at the discretion of the Council, for the best paper by a Student on heavier-than-air craft or any analogous subject. Its value is five guineas.

Usborne Prize

The Usborne Prize is offered annually, at the discretion of the Council, for the best paper by a Student on some subject in connection with Aero Engines. Its value is five guineas.

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The Major Baden-Powell Memorial Prize is awarded in May and December of each year to the student who is considered the best student by the examiners in the Society's Association Fellowship examinations. Its value is three guineas.

Elliott Memorial Prize

The Elliott Memorial Prize is awarded twice yearly to the apprentice at Halton who has the highest percentage of marks in the passing-out examination. Its value is 2½ guineas for each award.

R. P. Alston Memorial Prize

The R. P. Alston Memorial Prize is awarded to any Graduate or Student of the Society for work done leading to improvement in the safety of aircraft, and particularly for improvement in stability and control. Its value is approximately £5.

Branch Prize

The Council offer an annual prize of twenty guineas for the best paper read before the Branches during the previous lecture Session. The prize is open to any member of the Society or of any Branch.

Journal Premium Awards

The Council has set aside an annual sum of £250 to be given as premium awards to authors of papers published in the Journal.

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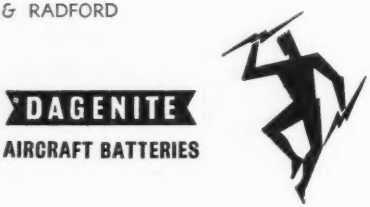

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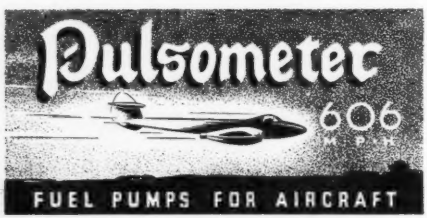

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Parnall Aircraft Ltd.	8 South Street, Park Lane, W.1	Grosvenor 2771-2
Percival Aircraft Ltd.	Luton Airport, Luton, Beds.	Luton 2960
Peto & Radford	Chequers Lane, Dagenham, Essex	Rainham 34
Philidas Ltd.	The Aerodrome, Reading, Berkshire	Wargrave 218
Sir Isaac Pitman & Sons Ltd.	Parker Street, Kingsway, W.C.2	Holborn 9791
Plastics	Temple Press, Bowling Green Lane, E.C.1	Terminus 3636
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Rolls-Royce Ltd.	Derby	Derby 2424
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Serck Radiators Ltd.	Warwick Road, Birmingham, 11	Victoria 0531
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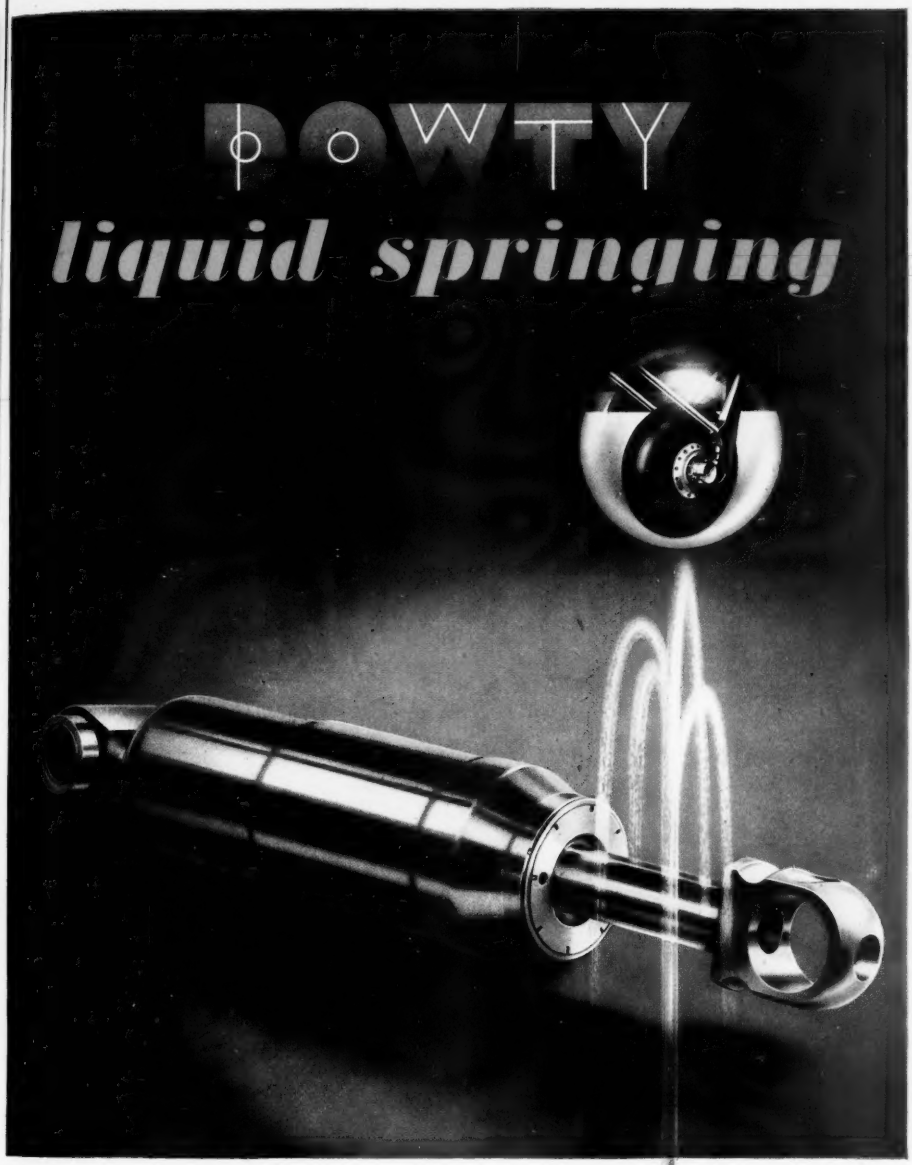

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Printed by the Lewes Press (Wightman & Co., Ltd.), Friars Walk, Lewes, England, and Published by the Royal Aeronautical Society, 4, Hamilton Place, London, W.1, England.

The Journal of THE ROYAL AERONAUTICAL SOCIETY

MARCH 1946

NO. 423 VOL 50